10^a/R

F. W. Harden.

INTERNATIONAL LIBRARY OF TECHNOLOGY

A SERIES OF TEXTBOOKS FOR PERSONS ENGAGED IN THE ENGINEERING PROFESSIONS AND TRADES OR FOR THOSE WHO DESIRE INFORMATION CONCERNING THEM. FULLY ILLUSTRATED AND CONTAINING NUMEROUS PRACTICAL EXAMPLES AND THEIR SOLUTIONS

STEAM BOILERS
STEAM PUMPS

SCRANTON:
INTERNATIONAL TEXTBOOK COMPANY

7

lt7a

BURR PRINTING HOUSE,
FRANKFORT AND JACOB STREETS,
NEW YORK.

PREFACE

The International Library of Technology is the outgrowth of a large and increasing demand that has arisen for the Reference Libraries of the International Correspondence Schools on the part of those who are not students of the Schools. As the volumes composing this Library are all printed from the same plates used in printing the Reference Libraries above mentioned, a few words are necessary regarding the scope and purpose of the instruction imparted to the students of—and the class of students taught by—these Schools, in order to afford a clear understanding of their salient and unique features.

The only requirement for admission to any of the courses offered by the International Correspondence Schools, is that the applicant shall be able to read the English language and to write it sufficiently well to make his written answers to the questions asked him intelligible. Each course is complete in itself, and no textbooks are required other than those prepared by the Schools for the particular course selected. The students themselves are from every class, trade, and profession and from every country; they are, almost without exception, busily engaged in some vocation, and can spare but little time for study, and that usually outside of their regular working hours. The information desired is such as can be immediately applied in practice, so that the student may be enabled to exchange his present vocation for a more congenial one, or to rise to a higher level in the one he now pursues. Furthermore, he wishes to obtain a good working knowledge of the subjects treated in the shortest time and in the most direct manner possible.

In meeting these requirements, we have produced a set of books that in many respects, and particularly in the general plan followed, are absolutely unique. In the majority of subjects treated the knowledge of mathematics required is limited to the simplest principles of arithmetic and mensuration, and in no case is any greater knowledge of mathematics needed than the simplest elementary principles of algebra, geometry, and trigonometry, with a thorough, practical acquaintance with the use of the logarithmic table. To effect this result, derivations of rules and formulas are omitted, but thorough and complete instructions are given regarding how, when, and under what circumstances any particular rule, formula, or process should be applied; and whenever possible one or more examples, such as would be likely to arise in actual practice—together with their solutions—are given to illustrate and explain its application.

In preparing these textbooks, it has been our constant endeavor to view the matter from the student's standpoint, and to try and anticipate everything that would cause him trouble. The utmost pains have been taken to avoid and correct any and all ambiguous expressions—both those due to faulty rhetoric and those due to insufficiency of statement or explanation. As the best way to make a statement, explanation, or description clear is to give a picture or a diagram in connection with it, illustrations have been used almost without limit. The illustrations have in all cases been adapted to the requirements of the text, and projections and sections or outline, partially shaded, or full-shaded perspectives have been used, according to which will best produce the desired results. Half-tones have been used rather sparingly, except in those cases where the general effect is desired rather than the actual details.

It is obvious that books prepared along the lines mentioned must not only be clear and concise beyond anything heretofore attempted, but they must also possess unequaled value for reference purposes. They not only give the maximum of information in a minimum space, but this information is so ingeniously arranged and correlated, and the

indexes are so full and complete, that it can at once be made available to the reader. The numerous examples and explanatory remarks, together with the absence of long demonstrations and abstruse mathematical calculations, are of great assistance in helping one to select the proper formula, method, or process and in teaching him how and when it should be used.

The subjects treated in the present volume are the construction, operation, care, and management of steam boilers and steam pumps, together with their various adjuncts and accessories. The aim has been to produce a work that will be of direct value to the man who looks after the machinery about a steam plant, and to the engineer and fireman. While not written for that purpose, this volume should prove of considerable value to designers, especially to those who have not actually fired a boiler or looked after the operating of pumps. The treatment accorded to the various subjects is amply sufficient to enable a fireman or engineer to pass any of the usual examinations preliminary to the granting of a license.

The method of numbering the pages, cuts, articles, etc. is such that each subject or part, when the subject is divided into two or more parts, is complete in itself; hence, in order to make the index intelligible, it was necessary to give each subject or part a number. This number is placed at the top of each page, on the headline, opposite the page number; and to distinguish it from the page number it is preceded by the printer's section mark (§). Consequently, a reference such as § 16, page 26, will be readily found by looking along the inside edges of the headlines until § 16 is found, and then through § 16 until page 26 is found.

INTERNATIONAL TEXTBOOK COMPANY.

CONTENTS

TYPES OF STEAM BOILERS.

FIRE-TUBE BOILERS.

1. A **steam boiler** is an apparatus whose duty it is to generate steam for power or heating purposes.

2. A boiler consists of the following essential parts: A **furnace** in which the combustion of the fuel takes place; a **vessel** to contain the water to be evaporated; a **steam space** to contain the generated steam; a **heating surface** to transmit the heat of the furnace to the water; a **chimney** to carry away the products of combustion and to give a draft to the fire; various **attachments,** or **fittings,** to feed the boiler with water, to carry away the generated steam, to indicate the pressure of the steam, etc.

3. Boilers are built in a variety of forms to meet the varying requirements of different classes of work. They may be roughly divided into three classes: **stationary, locomotive,** and **marine boilers.**

4. The **plain cylindrical boiler** is shown in Figs. 1, 2, and 3. It consists essentially of a long cylinder called the **shell.** This shell is made of iron or steel plates riveted together as shown in Fig. 1. The ends of the cylinder are closed by flat or hemispherical plates called the **heads** of the boiler. In Fig. 2 the front head is shown carrying the fittings B, C, C_1, and C_2. In this type of boiler the heads are

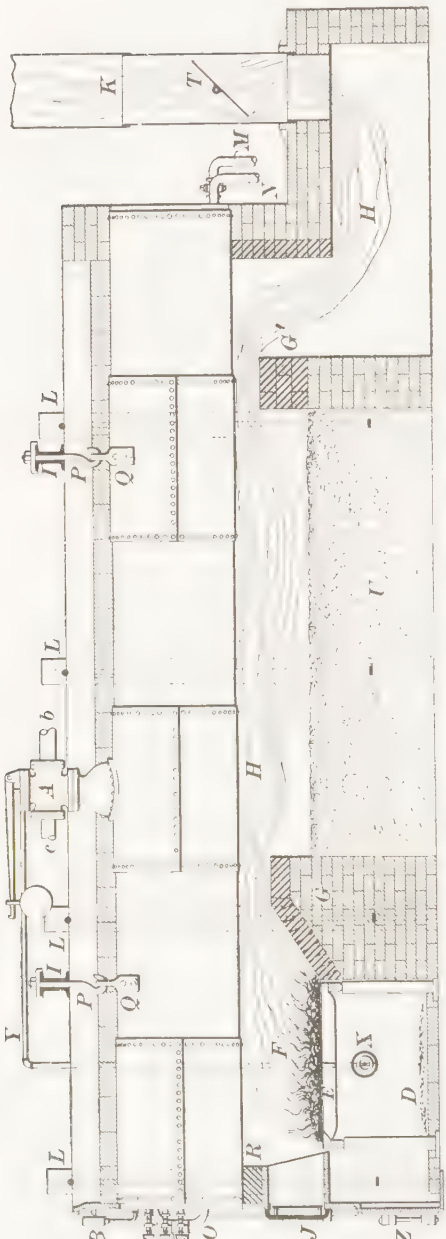


FIG. 1.

often made of thick cast iron, though wrought-iron plate may be used. The hemispherical, or dished, form of head is stronger than the flat head, and is therefore generally used for this type of boiler.

The boiler is enclosed by side walls of brick. The channel beams *I*, *I* are laid across these brick side walls, and the boiler is suspended from these beams by means of the hooks *P*, *P* and eyes *Q*, *Q* (see Figs. 1 and 3), the latter being riveted to the shell. The side walls are supported and kept from buckling by the binders, or buckstaves, *L*, *L* that are bolted together at the top and the bottom. The buckstaves are cast-iron bars of **T** section, as shown in the figure. The eyes *Q*, *Q* are placed about one-fourth the length of the shell from each

end. This method of suspending the shell allows it to expand and contract freely when heated or cooled.

The rear end of the shell is enclosed by the rear wall, as is shown in Fig. 1; the wall is continued back, forming the chamber *H*, into which the chimney, or stack, *K* opens. The front of the boiler, shown in Fig. 2, is of cast iron. The front end of the shell touches the firebrick lining *R*, and its weight comes upon the hooks *P*, *P*, the rear wall and firebrick lining *R* simply keeping it in position.

The furnace *F* is placed under the front end of the boiler shell. The fuel is thrown in through the furnace door *J*

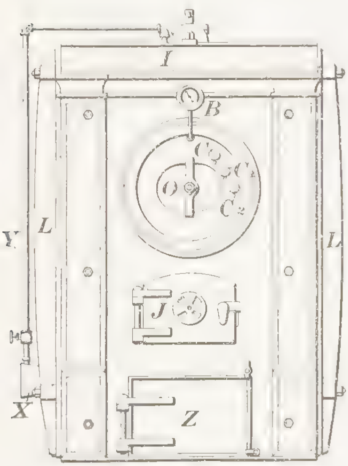


FIG. 2.

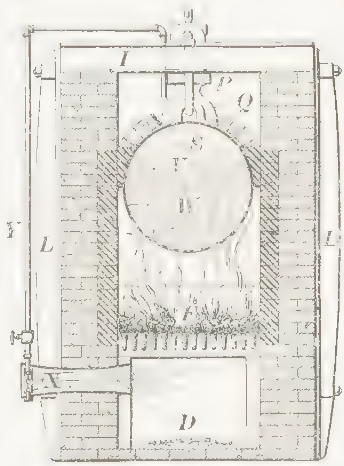


FIG. 3.

and burns upon the grate *E*, the ashes falling through the grate into the ash-pit *D*. To insure a supply of air sufficient for the complete combustion of the fuel, the furnace is sometimes supplied with a blower *X*. This consists of a cylinder leading into the ash-pit *D*, into which is led a jet of steam through the pipe *Y*. The jet escapes into the ash-pit with great velocity and carries with it a quantity of air. The air thus carried in is forced through the spaces between the grate bars into the burning fuel, thus producing rapid and complete combustion.

5. Behind the furnace is built the brick wall *G*, called the **bridge.** It serves to keep the hot gases in close contact with the under side of the boiler shell. As boilers of this type are generally quite long, a second bridge *G'* is usually added. The gases arising from the combustion of the fuel flow over the bridges *G* and *G'* into the chamber *H* and escape through the chimney *K*. The velocity of flow of the gases, and hence the intensity of the fire, is regulated by the damper *T* that is placed within the chimney. The space *U* between the bridges is filled with ashes or some other good non-conductor of heat. The door *Z* in the boiler front gives access to the ash-pit for the removal of the ashes. The tops of the bridges, the inner surface of the side walls and rear wall, and, in general, all portions of the brickwork exposed to the direct action of the hot gases are made of firebrick (shown in Figs. 1 and 3 by the dark section lining) since this material is able to withstand a very high temperature.

It will be seen by referring to Fig. 3 that the upper portion of the boiler shell is covered with firebrick in such a manner as to prevent the hot gases coming into contact with the shell above the water-line *V*. It is a general rule in the construction and setting of fire-tube boilers that *under no circumstances should the fire-line be carried above the water-line*, for if the hot gases come in contact with the part of the boiler above the level of the water, it will become unduly heated, and thus weakened, and will be liable to rupture. In fact, a great number of boiler explosions are caused by "low water" in the boiler, in which condition the water-line is below the fire-line.

The top of the shell is covered by brickwork or some other non-conducting material to prevent radiation of heat. The boiler is filled with water through the feedpipe *N*, which leads to a pump or injector. When in operation, the water stands at about the level *V*, the space *S* above being occupied by the generated steam. The **safety valve** is shown at *A*. By means of this attachment the steam pressure is kept from rising above the desired point. The pipe *b* is the main

steam pipe leading to the engine; the pipe *c* provides for the escape of the waste steam when the safety valve blows off.

The steam gauge *B* indicates the pressure of the steam in the boiler. The gauge is attached to a pipe that passes through the front head into the steam space.

The gauge-cocks C, C_1, and C_2 are placed in the front head of the shell; they are used to determine the water level. For instance, if the cock C_1 is opened and water escapes, it is evident that the water-line is above the cock C_1, while if steam escapes, the level must be below C_1.

The manhole *O* is simply a hole placed in the front head through which a man may enter and inspect or clean the boiler. The hole is closed by a plate and yoke.

To permit the boiler to be emptied, it is provided with a blow-off pipe *M*, through which the water or sediment may be discharged.

These boilers are made from 30 to 42 inches in diameter and from 20 to 40 feet long, though in rare instances they have been constructed with a diameter of 48 or more inches and a length of 60 or even 100 feet.

6. Plain cylindrical boilers are much used in mining districts, where fuel is very cheap; on account of their small water-heating surface, they are very uneconomical and hence are not generally used where fuel is expensive. The advantages of this boiler are: Cheapness of construction, strength, durability, and ease of access for cleaning and repairs.

7. The **flue boiler** differs from the plain cylindrical boiler in having one or more large flues running lengthwise through the boiler shell below the water-line.

Such a boiler is shown in elevation and section in Figs. 4, 5, and 6.

The flues *A*, *A* are fixed at the ends in the front and rear heads of the shell, respectively. The front end of the shell is prolonged, forming the **smokebox** *B*, into which opens the smokestack *C*. The front of the smokebox is provided with a door *E*. The boiler shell is also provided with the

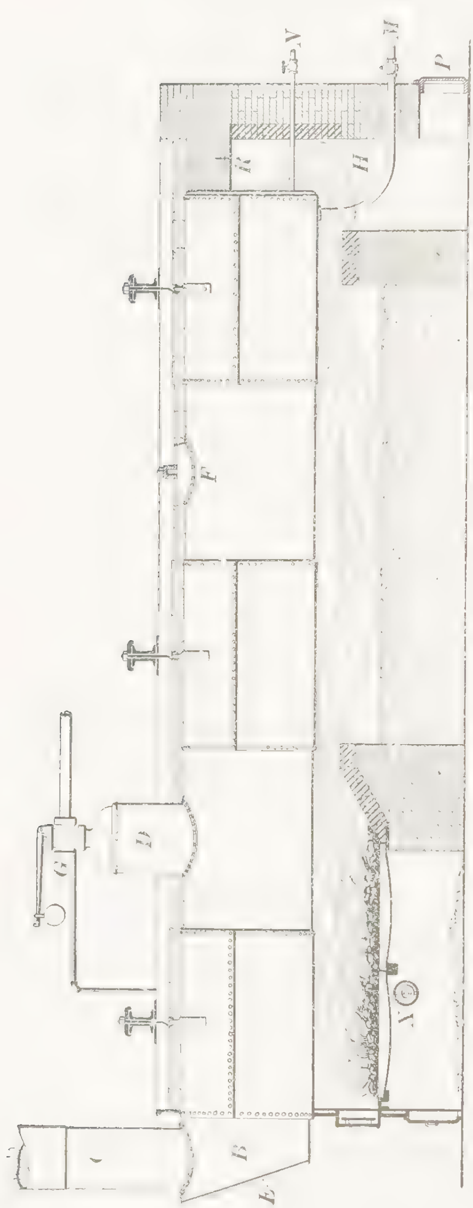


FIG. 4.

dome *D*, which forms a chamber where steam may collect and free itself from its entrained water before passing to the engine. The hot gases pass over the bridges to the chamber *H* and then back to the front end through the flues *A*, *A* into the smokebox *B*, and out through the stack *C*. It is plain that the water-heating surface is increased over that of the plain cylindrical boiler by the cylindrical surface of the flues *A*, *A*. In other particulars the description of the plain cylindrical boiler applies equally well to the flue boiler.

8. The Return-Tubular Boiler. This type of boiler is a development of the flue boiler, the two large flues

of the latter being replaced by a large number of small tubes. The object of introducing the numerous tubes is to increase the heating surface of the boiler.

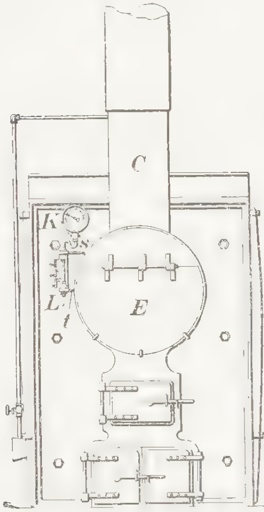


FIG. 5

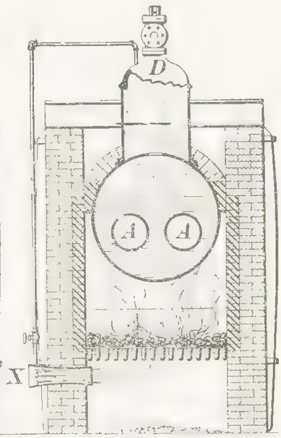


FIG. 6.

A side view of a tubular boiler is shown in Fig. 7; a cross-section through the tubes is shown in Fig. 8. The

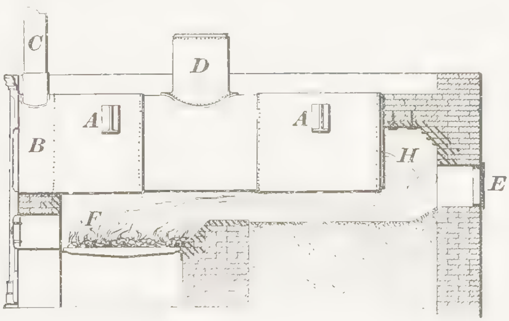


FIG. 7.

tubes extend the whole length of the shell; the ends are beaded into holes in the heads of the boiler. The front end

of the shell projects beyond the head, forming the smokebox *B*, into which opens the stack *C*.

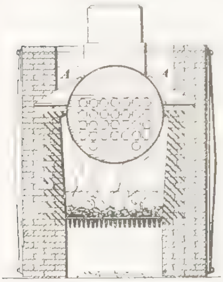


FIG. 8.

The shell is suspended on the side walls by the brackets *A*, *A*, which are riveted to the shell. The boiler is usually provided with a dome *D*, though this is sometimes left off. The walls are built and supported by buckstaves in practically the same manner as those previously described. Since this type of boiler is generally short, one bridge only is used. Firebrick is used for all parts of the wall exposed to the fire or heated gases. The fittings are not shown in the figure. The safety valve would be placed on top of the dome, and the pressure gauge and gauge-cocks would be placed on the front. The manhole is either in one of the heads or on top of the shell. The feedpipe may enter the front head or the top, while the blow-off pipe is placed at the bottom of the shell, at the rear end. Access is given to the rear end of the boiler through the door *E*.

As usual, the furnace *F* is placed under the front end of the boiler. The gases pass over the bridge, under the boiler into the chamber *H*, then back through the tubes to the smokebox *B*, and out of the stack *C*.

The return-tubular boiler is probably used in the United States more than any other. The details of its construction and setting will be shown later.

9. Cornish and Lancashire Boilers.—In the three forms of boilers considered so far, the furnace is placed outside the shell of the boiler; such boilers are said to be **externally fired.** On the invention of the single-flue boiler, the idea was conceived of placing the fire in the flue, and the result is the so-called **Cornish boiler,** a cross-section of which is shown in Fig. 9.

The boiler is set in masonry in such a manner as to form the passages *A*, *A*, and *B*. The grate is supported in the

single large flue *C*. The heated gases pass from the furnace to the rear through the flue *C*, they then return beneath the boiler through the flue *B*, and finally return to the rear through the side flues *A*, *A*, and thence out of the chimney. This path of the gases constitutes the **split draft.**

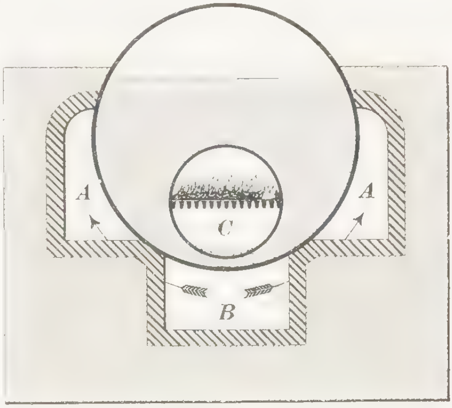


FIG. 9.

It was formerly the general practice to arrange the brick-work setting so that the gases returned to the front through the side flues *A*, *A* and returned to the rear through the lower flue *B*. It was found, however, that this practice retarded the circulation of the water and rendered the shell more liable to strains due to unequal expansion and contraction. Consequently, the first method of producing the split draft is used almost exclusively in modern practice.

As shown in the figure, the brickwork passages are lined with firebrick.

10. The **Lancashire boiler** is a modification of the Cornish type. In order to give a large grate area and a large heating surface for the same diameter of shell, two large furnace flues are substituted for the one flue of the Cornish type. The brickwork setting (Fig. 10) is precisely similar to that of the Cornish boiler, Fig. 9, and the split draft is formed in the same manner.

The large furnace flues of internally fired boilers, of which the Cornish and Lancashire are examples, are subjected to an external collapsing pressure equal to the pressure of the steam. The greater the diameter of the flue, the

more liable it is to collapse; consequently, the Lancashire possesses an advantage over the Cornish type in this respect, since each of its two flues is necessarily of smaller diameter than the single flue of the Cornish boiler. Various measures are taken to strengthen the furnace flues of internally fired boilers. They are sometimes made corrugated; again, they are strengthened by having channel irons riveted around them. A very common method, however, is to stiffen them by transverse conical water legs, as shown in Fig. 11. The water legs not only stiffen the flue, but also provide an opportunity for the circulation of the water and split up the gases on their way through the flue, thereby providing an increased heating surface.

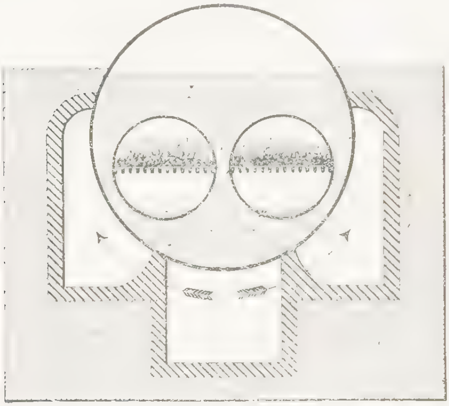


FIG. 10.

11. The **Galloway boiler** is a sort of combination of the Cornish and Lancashire types. It has two internal furnace flues fitted with grates, ashpit, etc. in the usual manner. Instead of extending through the whole length of the shell, the two flues unite just behind the bridge into one large kidney-shaped flue that extends from this junction to the rear head of the shell. This large flue is strengthened by a large number of water legs of the form shown in Fig. 11. The setting of the Galloway boiler is similar to that shown in Figs. 10 and 11. The draft is split as previously described.

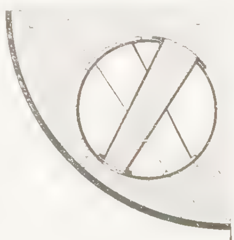


FIG. 11.

12. The Cornish, Lancashire, and Galloway boilers belong to the general class known as **internally fired** boilers. The chief objection to these three types of boilers is the liability of the internal flue to collapse and the straining actions set up by the expansion and contraction of the flue. The chief point in favor of these boilers and in favor of internally fired boilers generally is their economy in the use of fuel. Generally speaking, all conditions being the same, an internally fired boiler is 10 per cent. more economical than an externally fired boiler. This fact is due to the loss of heat by radiation through the brickwork setting of the latter class of boilers.

The three types of boilers just described are extremely popular in England and on the continent of Europe, but they are little used in America.

13. **The Clyde Boiler.**—A stationary boiler combining the features of the Lancashire and multitubular types is shown in Fig. 12. In consists of a large cylindrical shell *a*, the ends of which are closed by the flat heads *b*, *b*. A large furnace flue *c* of the corrugated type, known technically as the **Morison suspension furnace flue,** extends clear through the boiler and is securely riveted to the two heads, which are flanged inwards for this purpose. Above and beside the furnace flue and parallel thereto and below the water-line is a nest of tubes *d* that extend from head to head. The front ends of these tubes open into a smokebox *e* that connects with the chimney or stack *f*. The flat heads are stayed by through stayrods *g*, *g* in the steam space, which prevent deflection of the heads. The remaining parts of the flat heads are supported by the tubes, which are expanded and beaded over, and by the furnace flue.

The furnace is placed within the furnace flue and, as usual, consists of the grate *h*, the ash-pit *i*, and the bridge *k*. The gases of combustion flow to the rear into the combustion chamber *l* and then pass through the tubes to the front and into the smokebox. The combustion chamber is formed by a thin cylindrical shell attached to the rear end of the

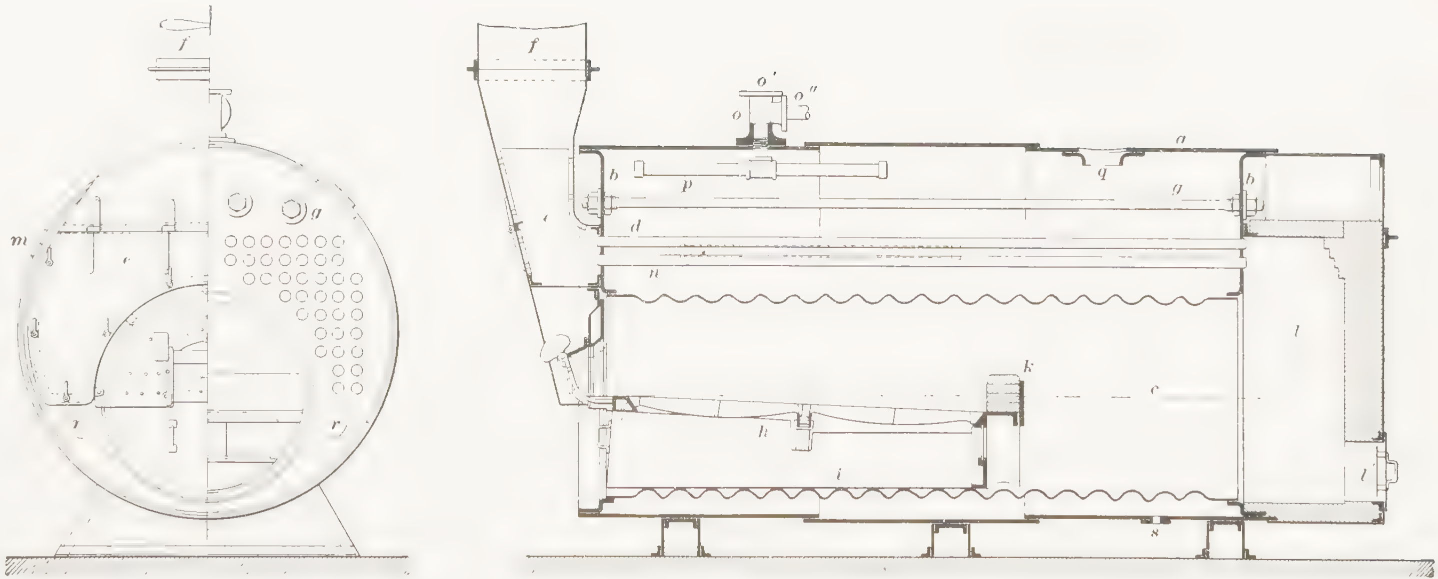


FIG 12

boiler, and is lined with firebrick, as shown. A door l' gives access to the combustion chamber for the removal of ashes and soot and for the purpose of examination and repair. This type of boiler evidently gives a very large amount of heating surface in proportion to the space it occupies.

The feedwater enters the boiler at m and, passing through the internal perforated feedpipe n, is discharged downwards alongside the shell in small streams. The various fittings are not shown in the illustration. The steam gauge and water column would naturally be located close to the front end of the boiler; the safety valve is intended to be bolted to the outlet o' and the steam pipe to the outlet o'' of the nozzle o. The steam is collected by the dry pipe p, which is perforated with numerous slots on top. The dry pipe is fairly effective in freeing the steam from any water that may be mixed with it. The manhole is at q and two handholes at r. The blow-off is attached at s. The boiler is entirely self-contained, i. e., it does not require any brickwork setting. It is simply bolted to three saddles that rest upon and are fastened to a suitable foundation.

A boiler of the kind just illustrated resembles the Scotch boiler used in marine work, and differs from it only in the fact that the combustion chamber is not surrounded by water. For this reason it is often called a **dry-back Scotch boiler,** although some engineers refer to it as the **Clyde** boiler, presumably because this type was originally designed in the shipyards of the Clyde, England.

14. The Locomotive, or Firebox, Boiler.—Next to the multitubular type, the firebox boiler is probably used more than any other type. It is used exclusively in railway service and also largely as a stationary boiler. A large proportion of the small portable combined engines and boilers used for agricultural purposes are of this type. The general construction is shown in Fig. 13. The shell is composed of two differently shaped parts riveted together. The front part of the shell is cylindrical; the rear part is usually of a rectangular cross-section with vertical sides or

of a trapezoidal section with inclined sides; in either case the top is semicylindrical. The furnace *F* is a box of the same shape as the rear end of the shell in which it is placed. A space is left between the sides and end of the furnace and the shell; this space is filled with water, as shown at *A*, *A*. A series of tubes extends from the front sheet of the furnace, or firebox, to the front head of the shell. The shell is prolonged beyond the front head, forming a smokebox *B*, into which opens the stack *C*.

As shown in this figure, the "water legs" (as the spaces *A*, *A* are called) extend only as far as the grate, the ash-pit *D* being formed in the brick setting. In many boilers of

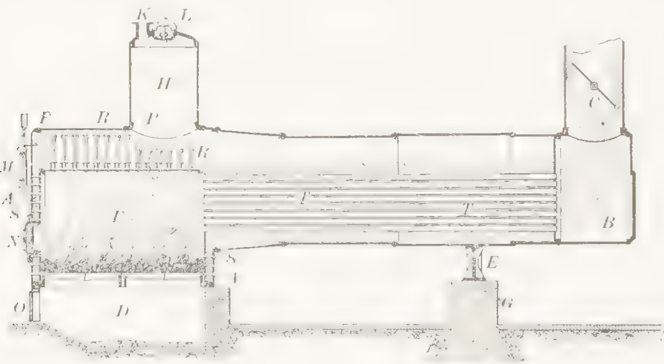


FIG. 13.

this type the water legs extend to the bottom of the ash-pit, and sometimes there is a water space below the ash-pit; that is, the furnace and ash-pit are entirely surrounded by water and no brickwork setting is required.

The boiler is supported at the front end by the cast-iron cradle *E* that rests upon the masonry foundation *G*. The rear end is supported upon a brick wall, which also forms the ash-pit. The boiler is usually provided with a dome *H*, from which is led the main steam pipe, which is bolted on at *K*. In the figure, the dome is provided with a manhole *L*. The feedwater may be introduced at any convenient point in the shell. The pressure gauge, water glass, and gauge-cocks are attached to the column *M*, which is placed in

communication with the interior of the shell. The furnace and ash-pit doors are shown at *N* and *O*, respectively. The safety valve is usually attached to the dome.

Since the flat sides of the furnace and shell are liable to bulge on account of the pressure, they must be braced or stayed. This is accomplished by the staybolts *S*, *S*. The flat top of the firebox is strengthened by a series of parallel girders *P*, *P*. As an additional security, the girders are sometimes attached to the shell by the "sling stays" *R*, *R*.

The gases of combustion pass directly from the furnace through the tubes *T*, *T* to the smokebox *B* and out of the stack *C*. In locomotives, a strong draft is obtained by allowing the exhaust steam to discharge through the smokestack. The escaping steam carries along the air and the escaping gases in the smokebox *B*, thereby drawing a new supply of gases through the tubes *T*, *T* and a supply of air through the grate.

The tubes of the locomotive boiler are about 12 feet long, 2 inches in diameter, and are made of iron or steel. The tubes of stationary and portable boilers of this type are generally of larger diameter, as there is less demand for great quantities of steam.

The locomotive type of boiler evidently is self-contained.

15. The Vertical Boiler.—This type is essentially a modification of the locomotive type placed on end. A common form of vertical boiler is shown in Fig. 14. It consists of a vertical cylindrical shell, in the lower end of which is placed a firebox *F*. The lower rim of the firebox and the lower end of the shell are separated by a wrought-iron ring *k*, to which both are riveted, the rivets going through both plates and ring. The shell and firebox are also stayed together by the staybolts *a*, *a*. The space between the two is filled with water, so that the firebox is surrounded by it. The boiler shell and likewise the grate *E* rest upon a cast-iron base *D* that forms the ash-pit. A series of vertical tubes *t*, *t* extend from the top sheet of the firebox to the upper head of the shell. The tubes serve as stayrods and

strengthen the flat surfaces that they connect. The upper ends of the tubes open directly into the chimney or smoke-stack K. The gases from the furnace thus pass directly through the tubes and out of the stack.

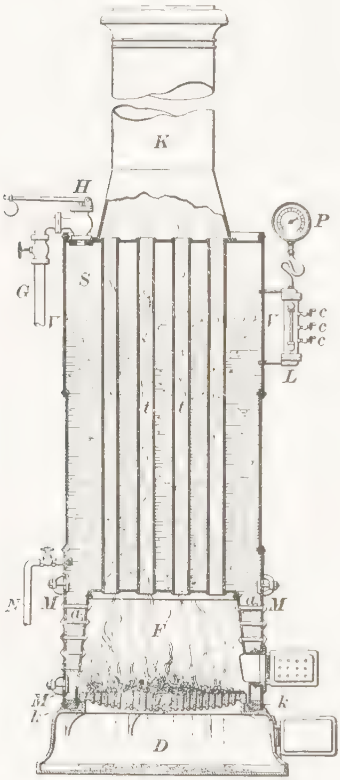


FIG. 14.

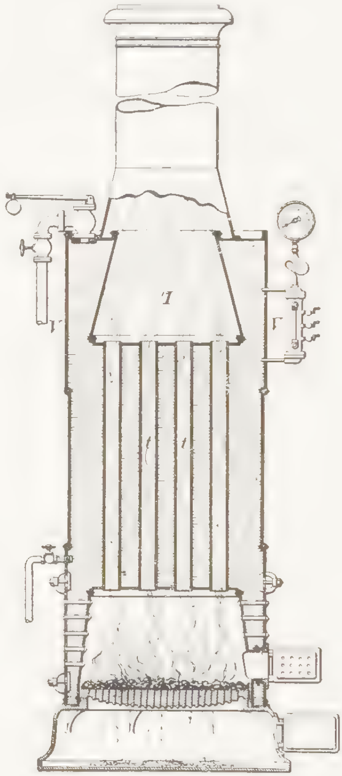


FIG. 15.

The safety valve is shown at H, with the main steam pipe G leading from it. The pressure gauge P and gauge-cocks c, c, c are attached to a column L that communicates in the usual manner with the interior of the shell. The construction of this type of boiler does not generally permit the use of manholes, but handholes M, M are placed in convenient positions for cleaning out mud and sediment.

16. When the tubes extend through the upper head of the boiler, as shown in Fig. 14, their upper ends pass through the steam space S above the water-line $V V$. This is looked upon as a bad feature, since the tubes are liable to become overheated and thus collapse when the boiler is forced.

In the form of vertical boiler shown in Fig. 15, this danger is avoided. A chamber, or smokebox, I extends from the upper head of the shell so that its bottom plate is always below the water-line. The upper ends of the tubes t, t are expanded into the lower plate of this chamber, and therefore the tubes are always surrounded by water from end to end. A vertical boiler constructed in this manner is said to have a **submerged head.** Aside from the submerged head, the construction of the boiler of Fig. 15 is similar to that of Fig. 14.

Vertical boilers are generally wasteful of fuel and are perhaps more liable to explosion than any other type. They are, however, self-contained, require but little floor space, and are easy to construct and repair. For these reasons the vertical type of boiler is very popular with a large class of steam users.

17. The tubular boilers so far described belong to a general class known as **fire-tube boilers,** and any boiler in which the flames and gases of combustion traverse the *inside* of the tube or flue belongs to this class.

WATER-TUBE AND SECTIONAL BOILERS.

18. Of late years, a type of boiler has come into extensive use in which the flames and gases of combustion are in contact with the *outside* of the tubes. The water is contained inside the tubes; hence, these boilers are known as **water-tube boilers.** Some of the leading types of water-tube boilers are described in the following articles.

19. The **Babcock and Wilcox water-tube boiler** is shown in Fig. 16. It consists essentially of a main horizontal drum *B* and of a series of inclined tubes *T*, *T*. (Only a single vertical row of tubes is shown by the figure, but it will be understood that each nest of tubes is composed of several vertical rows.) There are usually seven or eight of these vertical rows to each horizontal drum. The front ends of the tubes of a vertical row are all expanded into a hollow **header** *H*. The rear ends are expanded into a similar header, and the front and rear headers are placed in

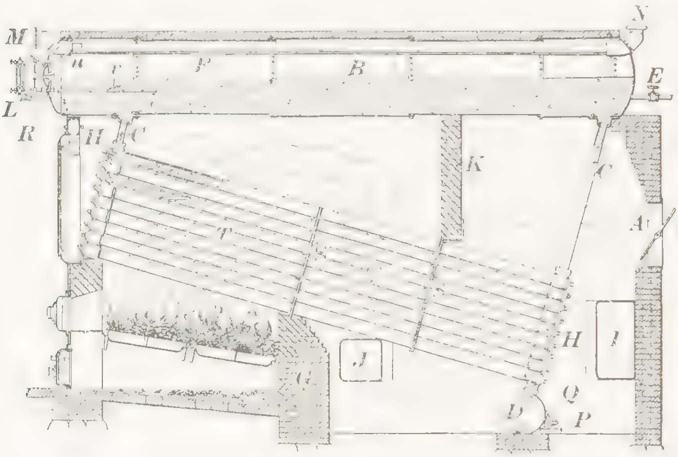


Fig. 16.

communication with the drum by tubes, or **risers**, *C* and *C*, respectively. In front of each tube, a handhole is placed in the header for the purpose of cleaning, inspecting, or removing the tubes.

The method of supporting the boiler is not shown in the figure. The usual method is to hang the boiler from wrought-iron girders resting on vertical iron columns. The brickwork setting is not depended on as a means of support. This make of boiler, in common with all others of the water-tube type, requires a brickwork setting to confine the furnace gases to their proper field.

20. The furnace is of the usual form and is placed under the front end of the nest of tubes. The bridge wall *G* is built up to the bottom row of tubes; another firebrick wall *K* is built between the top row of tubes and the drum. These walls and the baffle plates *S*, *S* force the hot furnace gases to follow a zigzag path back and forth between the tubes. The gases finally pass through the opening *A* in the rear of the wall into the chimney flue.

The feedwater is introduced through the feedpipe *E*. The steam is collected in the dry pipe *F*, which terminates in the nozzles *M* and *N*, to one of which is attached the main steam pipe and to the other the safety valve.

The pressure gauge, cocks, etc. are attached to the column that communicates with the interior of the shell by the small pipes *u* and *v*, the former of which extends into the dry pipe, the latter into the water.

At the bottom of the rear row of headers is placed the mud drum *D*. Since this drum is the lowest point of the water space, most of the sediment naturally collects there. This sediment may be blown out from time to time through the blow-off pipe *P*. The drum *D* is provided with a handhole *Q*. A manhole *R* is placed in the front head of the drum *B*. The heads of the drums are of hemispherical form and therefore do not require bracing. Access may be had to the space within the walls through the doors *I* and *J*.

21. The circulation of water takes place as follows: The cold water is introduced into the rear of the boiler; the furnace being under the higher end of the tubes, the water in that end expands upon being heated, and is also partially changed to steam; hence, a column of mingled water and steam rises through the front headers to the front end of the drum *B*, where the steam escapes from the surface of the water. In the meantime, the cold water fed into the rear of the drum descends to the rear headers through the long tubes *C* to take the place of the water that has risen in front. Thus, there is a continuous circulation in one direction, sweeping the steam to the surface as fast as it is

formed and supplying its place with cold water. Most of the sediment sinks to the mud drum D, from which it is blown out from time to time.

22. The **Root water-tube boiler** is shown in Figs. 17 and 18, the latter being an end view with the brickwork removed so as to show the various drums and connections. The construction of this boiler is very similar to that of the one just described. There is a nest of inclined tubes, the ends of which are expanded into cast-iron headers. The headers are placed in communication by the **U**-shaped return

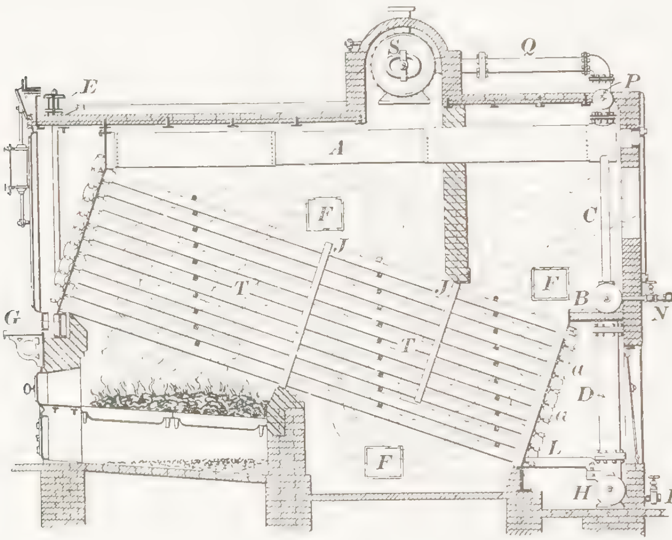


Fig. 17.

bends a, a. A continuous channel is therefore provided for the circulation of the water through the headers. There is a horizontal overhead drum A for each vertical section of tubes. These drums A, A are placed in communication with the transverse drum B by the tubes C, C. The drum B is in turn connected with the lower drum H by the two large water legs D, D. Finally the drum H communicates with the rear headers through the tubes L, L. There is thus an open circuit through the tubes T, drum A, tubes C, drum B,

water legs *D*, drum *H*, and tubes *L*. The water-line is at about the middle of the drums *A*, *A*, and the steam arising from the surface of the water first passes into the drum *P* and then into the main steam drum *S* through the pipes *Q*, *Q*. The main steam pipe, the safety valve, and other fittings may be attached to the drum *S* at the nozzles *U*, *V*, and *W*.

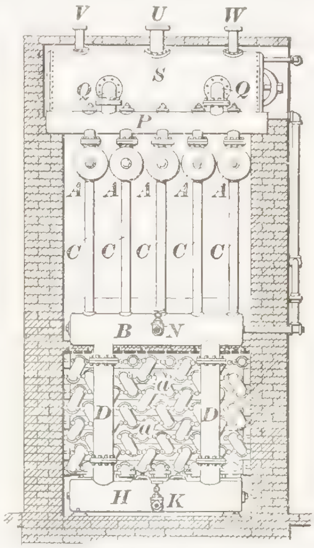


FIG. 18.

The feedwater is introduced into the drum *B* through the feed-pipe *N*. The circulation takes place in the same manner as in the boiler previously described. The drum *H* acts as the mud drum, being at the lowest point of the water circuit. The sediment may be blown out through the pipe *K*.

Access may be had to the interior of the setting through the doors *F*, *F*. The steam drum is provided with a manhole. The rear end of the boiler is supported by the brickwork foundation; the front end is supported by a beam *G* hung from the **I** beam *E*.

The arrangement of the bridge and baffle plates *J*, *J* and the course of the heated gases are precisely the same as in the Babcock and Wilcox boiler.

23. The **Heine water-tube boiler,** shown in Fig. 19, differs in many respects from those already described. It consists of a large main drum *A* that is above and parallel with the nest of tubes *T*, *T*. Both drum and tubes are inclined at an angle with the horizontal that brings the water level to about one-third the height of the drum in front and about two-thirds the height in the rear. The ends of the tubes are expanded into the large wrought-iron water legs *B*, *B*. These legs are flanged and riveted to the shell,

which is cut out for about one-fourth its circumference to receive them, the opening being from 60 to 90 per cent. of the cross-sectional area of the tubes. The drum heads are of a hemispherical form and therefore do not need bracing.

The water legs form the natural support of the boiler, the front water leg being placed on a pair of cast-iron columns E that form part of the boiler front, while the rear water leg rests on rollers (shown at F) that may move freely on a cast-iron plate bedded in the rear wall. These rollers allow the boiler to expand freely when heated.

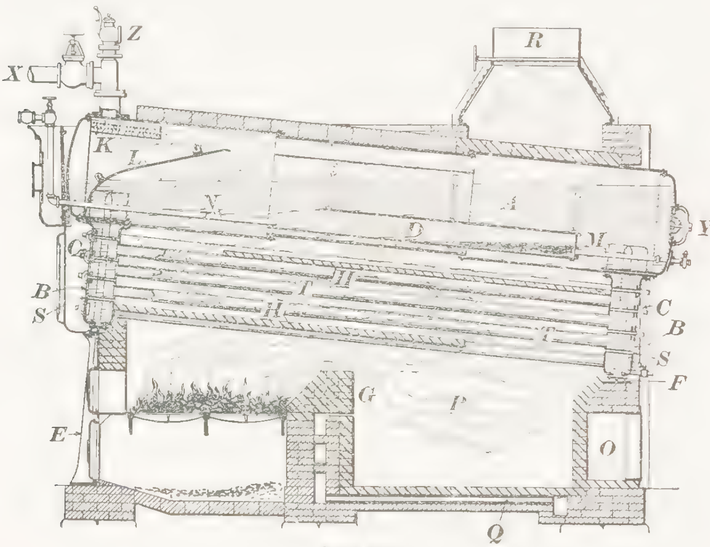


FIG. 29.

The boiler is enclosed by a brickwork setting in the usual manner. The bridge G, made largely of firebrick, is hollow, and has openings in the rear to allow air to pass into the chamber P and mix with the furnace gases. This air is drawn from the outside through the channel Q in the side wall and is, of course, heated in passing through the bridge. In the rear wall is the arched opening O that is closed by a door and is further protected by a thin wall of firebrick.

When it is necessary to enter the chamber P, the wall may be removed and afterwards replaced.

The feedwater is brought in through the feedpipe *N* which passes through the front head As the water enters, it flows into the mud drum *D*, which is suspended in the main drum below the water-line, and is thus completely submerged in the hottest water in the boiler. This high temperature is useful in precipitating the impurities contained in the feedwater, which settle in the mud drum *D* and may then be blown out through the blow-off pipe *M*.

Layers of firebrick *H*, *H* are laid at intervals along the rows of tubes and act as baffle plates, forcing the furnace gases to pass back and forth around the tubes. The gases finally escape through the chimney *R* placed above the rear end of the boiler. To protect the steam space of the drum from the action of the hot gases, the drum in the vicinity of the chimney is protected by firebrick, as shown in the figure.

The steam is collected and freed from water by the perforated dry pipe *K*. The main steam pipe, with its stop-valve, is shown at *X*, the safety valve at *Z*. In order to prevent a combined spray of mixed water and steam spurting from the front header and entering the dry pipe, a deflecting plate *L* is placed in the front end of the drum.

A manhole *Y* is placed in the rear head of the drum. The flat sides of the water legs are stayed together by the staybolts *S*, *S*, which are made hollow so as to give access to the outside of the tubes. In front of each tube is placed a handhole *C* that gives access to the interior of the tubes.

Where a battery of several of these boilers is used, an additional steam drum is placed above and at right angles to the drums *A*, *A*.

24. The **Stirling boiler,** shown in Fig. 20, is a departure from the regular type of water-tube boilers. It consists of a lower drum *A* connected with three upper drums *B*, *B*, *B* by three sets of nearly vertical tubes. These upper drums are connected by the curved tubes *C*, *C*, *C*. The curved forms of the different sets of tubes allow the different parts of the boiler to expand and contract freely without strain.

The boiler is enclosed, as shown, in a brickwork setting

that is provided with various holes *H*, *H*, so that the interior may be inspected or repaired. The boiler is suspended from a framework of wrought-iron girders not shown in the figure.

The bridge *E* is lined with firebrick and is built in contact with the lower drum *A* and the front nest of vertical tubes. An arch *D* is built above the furnace, and this, in

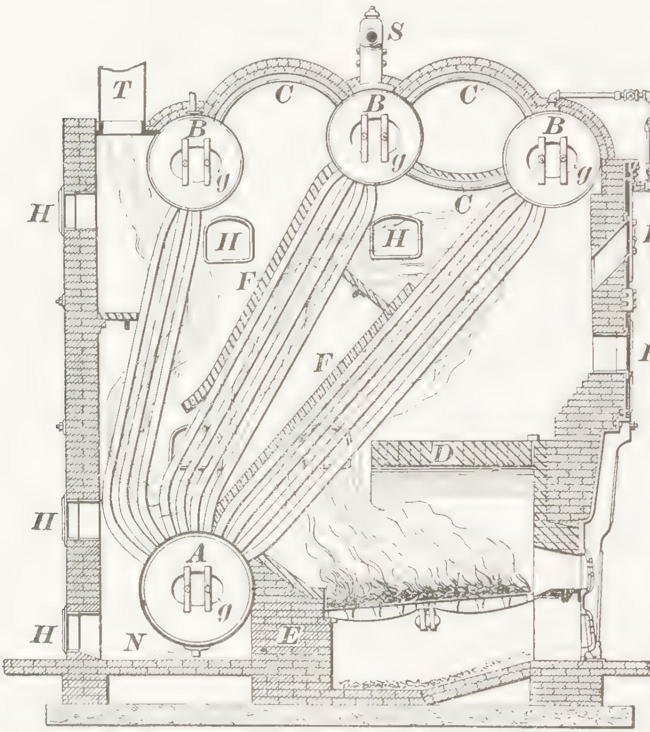


FIG. 20.

connection with the bafflers *F*, *F*, directs the course of the heated gases, causing them to pass up and down between the tubes. The arch and bafflers are made of firebrick.

The cold feedwater enters the rear upper drum and descends through the rear nest of tubes to the drum *A*, which acts as a mud drum and collects the sediment brought in by the water. A blow-off pipe *N* permits the removal of the

sediment. The steam collects in the upper drums *B*, *B*. The steam pipe and safety valve *S* are attached to the middle drum.

The chimney *T* is located behind the rear upper drum. Therefore, the cold feedwater enters the coolest part of the boiler, and the circulation of the water is directly opposite that of the escaping hot gases.

The water column *L*, with its fittings, is placed in communication with the front upper drum. All the drums are provided with large manholes *g*.

25. The **Hazelton** or **porcupine boiler,** shown in Fig. 21, is a vertical water-tube boiler of peculiar form. As shown in the figure, it consists of an upright cylinder *A*, into which are expanded a large number of radial tubes *B*, *B*, whose outer ends are closed, while the inner ends open into the cylinder *A*. The boiler is enclosed in a circular brickwork wall, on top of which is placed the chimney *H*, which is provided with a damper for regulating the draft. Below the tubes the wall is lined with firebrick and projects inwards, forming the furnace *E*. There are several fire-doors spaced equidistant around the circumference, one of which is shown at *K*. The grate *G* forms a ring between the central cylinder and the external brick wall. The space below the grate serves as the ash-pit. The water is

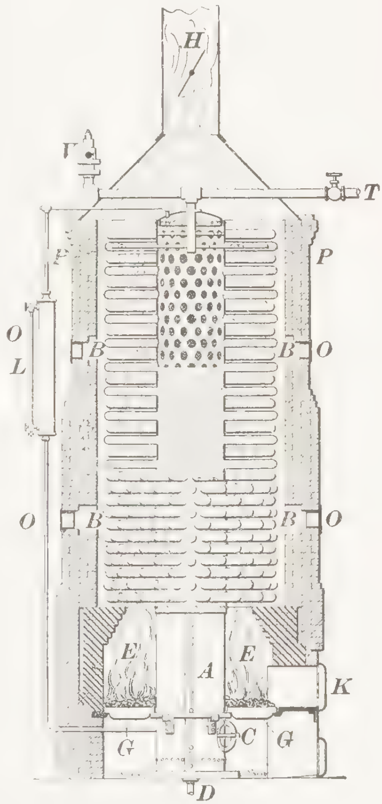


FIG. 21.

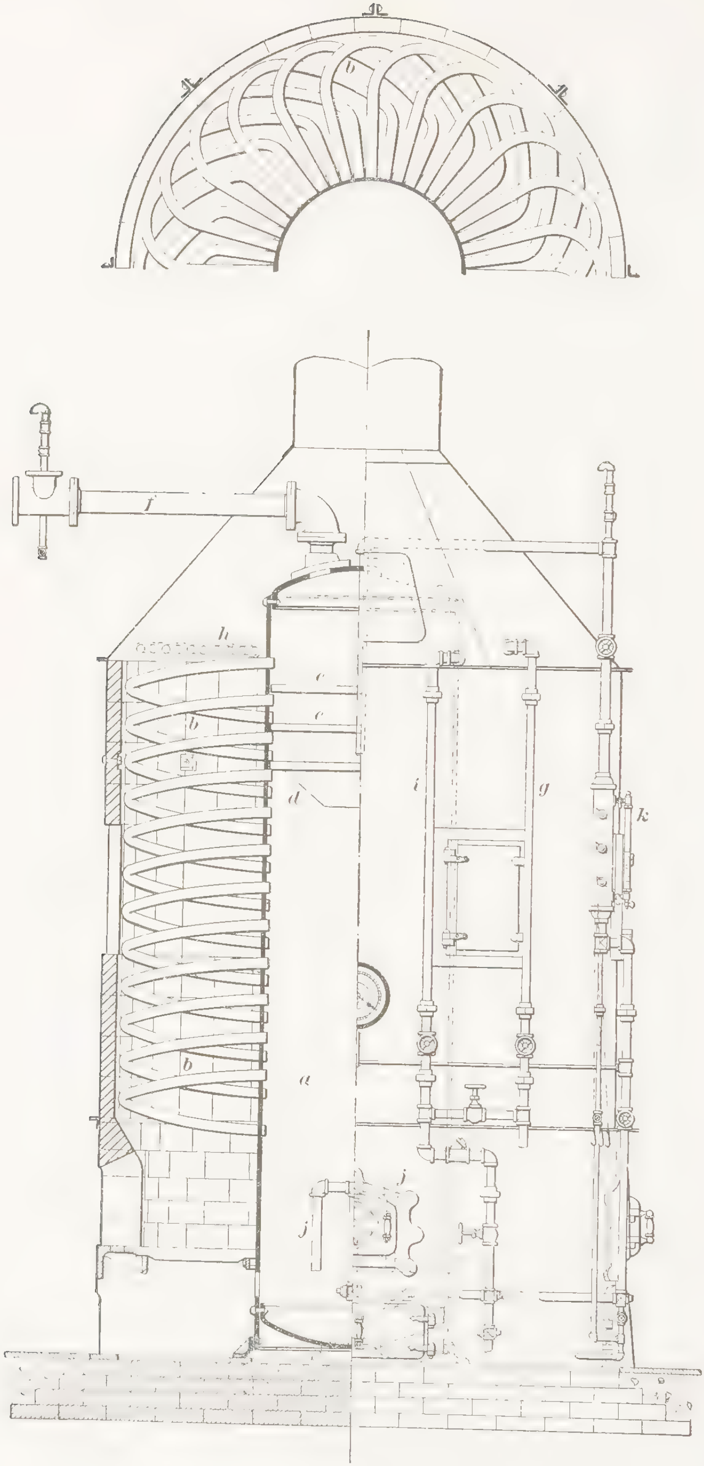


FIG. 22.

contained in the cylinder and tubes. The sediment naturally collects in the bottom of the cylinder and is blown out through the pipe *D* or removed through the manhole *C*. The steam is collected in the perforated dry pipe *P* and led to the main steam pipe *T*. The safety valve is shown at *V* and the water column at *L*. The openings *O*, *O* are left in the wall so that the interior may be inspected.

The heated gases pass from the furnace *E* between the tubes *B*, *B*, and by the time they have reached the chimney, the heat has been mostly absorbed by the water in the tubes.

This boiler does not require to be suspended in any way; the whole weight rests upon the foundation, which may be built in the ground.

26. The **Morrin "Climax"** boiler, shown in Fig. 22, is a water-tube boiler that somewhat resembles the porcupine boiler, and differs from it chiefly in that instead of radial tubes the standpipe or main shell *a* is fitted with a large number of loop-like tubes *b*, *b*, the ends of which are expanded into the shell. The furnace is circular, as in the porcupine boiler, and in order to give free access to the fire, four furnace doors are provided. A deflector plate *d* is fitted to the shell a little above the water level, which tends to throw back any water carried up by the steam. The upper portion of the central shell is divided by a series of diaphragms *c*, *c* into a series of superheating chambers, through which the steam is compelled to circulate successively by the connecting loop-like tubes. The steam thus becomes thoroughly dried and somewhat superheated before it enters the main steam pipe *f*. The feedwater coming from the pump is discharged through the delivery pipe *g* into a spiral feed-coil *h* resting on top of the tubes, where it is heated to a high temperature. It leaves the coil through the pipe *i* and passes downwards, finally being discharged into the bottom of the shell through the internal feedpipes *j*, *j*. The water column *k* is connected to the top and bottom of the central shell. The safety valve is attached to a **T** placed in the main steam pipe close to the boiler.

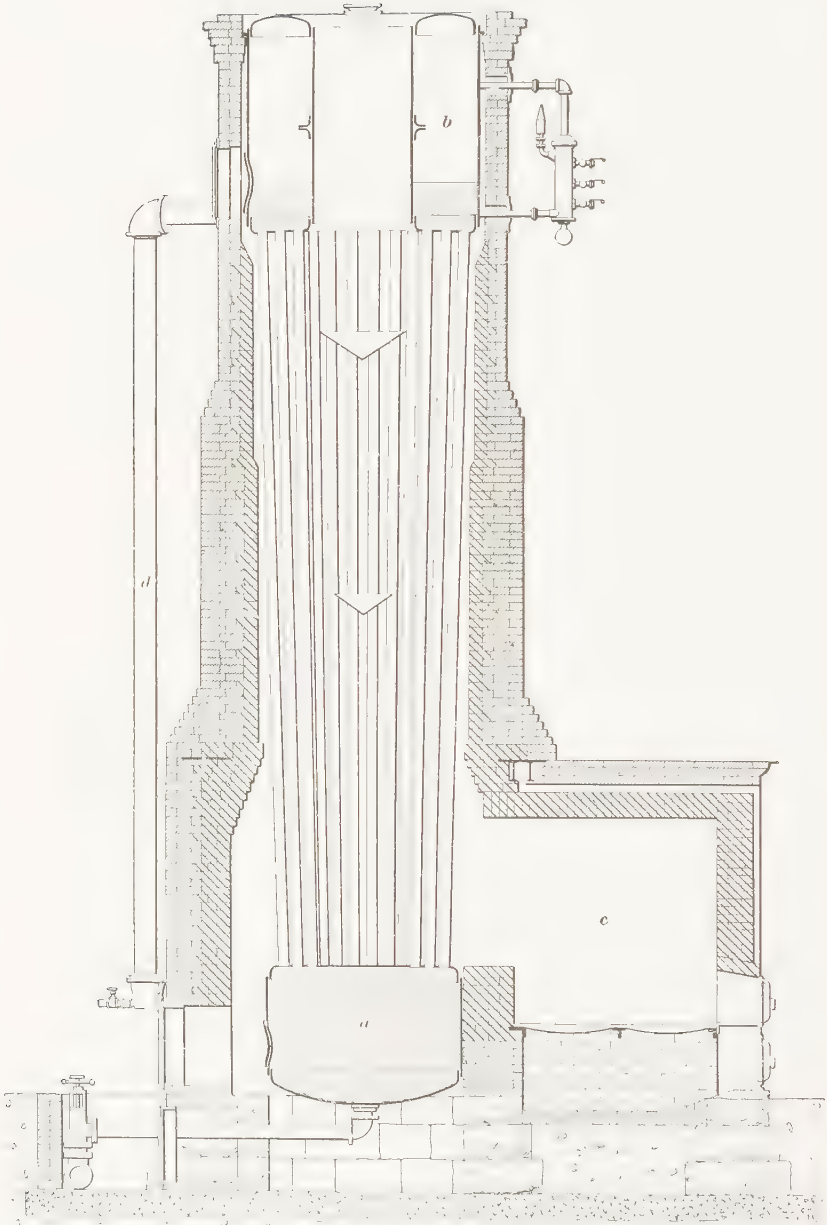


Fig. 23.

27. The **Cahall** boiler may aptly be called a **vertical** water-tube boiler. As a reference to Fig. 23 will show, it consists of a cylindrical mud drum *a* and steam drum *b*, which are connected by nearly vertical tubes. The furnace *c* is placed to one side of the boiler, and the gases of combustion surround the tubes and finally pass through a central passage in the steam drum to the smokestack. The steam becomes slightly superheated in this steam drum, through coming in contact with the surface of the central passage, which is kept at a fairly high temperature by the escaping gases. The steam drum and mud drum are connected by an external circulating pipe *d* that enters the steam drum some distance below the water-line. The feedwater enters the mud drum and, becoming highly heated, rises through the vertical tubes to the steam drum, where the steam bubbles are liberated. Some of the water in the lower part of the steam drum flows continually into the circulating pipe, and since this pipe is not exposed to the heat of the fire, the density of the water in it is much greater than the density of the water in the vertical boiler tubes. In consequence, the water is continually flowing downwards and a rapid circulation is promoted.

28. The **Harrison safety boiler,** shown in Fig. 24, is a sectional safety boiler, but not of the water-tube type. It is composed of hollow cast-iron or steel sections *A*, *A*, called **units,** that are accurately faced and bolted together. Each section is composed of two or more approximately spherical vessels, and the sections are bolted together in a zigzag manner, as shown in the figure, so as to form a solid slab. Each bolt runs from top to bottom through all the units, as shown at *C*. There are several of these vertical slabs of sections suspended side by side from the girders *B*, *B*.

The boiler is enclosed by a brickwork setting that is lined with firebrick. The top of the boiler is likewise covered with firebrick to prevent radiation. The wall is pierced with the openings *D*, *D* for the purpose of inspecting the interior. The bridge *G* and bafflers *H*, *H* direct the hot gases back and forth between the sections.

The feedwater enters the boiler at its lowest point through the feedpipe *N*. The steam pipe is bolted on at the flange *J*. The water column *L*, placed in front at the height of the water level, is connected by pipes with the steam and water spaces, respectively.

The chimney flue is placed at the rear near the floor; the draft is regulated by the damper *T*.

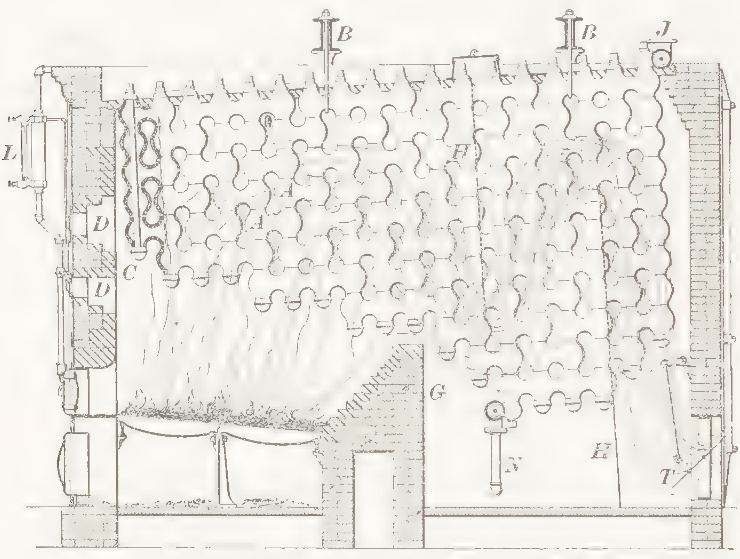


FIG. 24.

The boiler is essentially a safety boiler. If the steam pressure becomes excessive, the bolts *C* will elongate a little and allow steam to escape through the joints. Even if the pressure should be sufficient to burst a unit, there would be no disastrous explosion, and it would be only necessary to replace the unit.

29. Comparative Advantages of Water-Tube and Fire-Tube Boilers.—In all water-tube boilers, the whole or the greater part of the heating surface is formed by the

outer surface of the tubes of which the boiler is composed. The lower half of the steam drum, up to the fire-line, is also considered as heating surface.

The various water-tube boilers just described are coming into extensive use, some of the most important advantages claimed for boilers of this class being: (1) *Accessibility*, (2) *ease of repair*; (3) *small cost of repairs*; (4) *rapidity of steam generation without undue straining*; (5) *safety from disastrous explosions*; (6) *economy of fuel.*

Of these claims, the first five have been pretty well established as being correct. Regarding the sixth claim, it cannot be denied that in many instances fire-tube boilers have been replaced with water-tube boilers and a marked degree of saving of coal has been effected. On the other hand, ordinary fire-tube boilers, when properly designed, set, and operated, have often shown the same economy, and it is most likely that under equally advantageous conditions either class will show practically the same economy.

30. The safety from disastrous explosions is due to the small amount of water contained in the water-tube boiler, and also to the greater comparative strength of the tubes, owing to their small diameter. Instead of a large body of highly heated water being suddenly liberated and converted into steam, as is the case if a sheet of a fire-tube boiler ruptures, the splitting of one or even of a number of tubes of a water-tube boiler will only permit the gradual escape of a small body of water. While this may occasionally prove disastrous to the attendant, the damage to property and to persons in the vicinity is likely to be very small.

31. Disadvantages of Water-Tube Boilers.—Some of the objections raised against water-tube boilers are: (1) *The splitting of one tube will throw the boiler temporarily out of service*; (2) *difficulty of maintaining the water level.*

The first objection seems to be valid, since temporary repairs cannot be effected without shutting down. In a

fire-tube boiler a split tube can be temporarily repaired very quickly by plugging, and this can be done without throwing the boiler out of service.

Regarding the second objection, it is plain that as water-tube boilers contain but a limited quantity of water, it requires constant feeding to replace the water evaporated into steam. But as pumps and injectors can be regulated to a nicety to supply the required amount of water, the difficulty of maintaining a steady water level is not a serious one.

32. There is an endless variety of water-tube boilers, but the principles of operation are practically the same in all of them. After studying carefully the chapter on water-tube boilers, the student should experience no difficulty in understanding the operation of any water-tube boiler of which he may be placed in charge.

MISCELLANEOUS TYPES.

33. In addition to the types of boilers above described, there are many others embodying special features or of peculiar construction. Some of these are modifications of the types already described; others have nothing in common with them.

34. Of these miscellaneous types, the **Field boiler** may be mentioned. This boiler consists of a vertical cylindrical shell containing a plain flat-top firebox. A single large flue passes from the crown sheet of the firebox through the upper head of the boiler shell to the chimney. The remarkable feature of this boiler is the use of the so-called **Field tubes.** The construction of one of these tubes is shown in Fig. 25. It consists of two concentric tubes, the outer one being closed at one end and open at the other, which is expanded into the crown sheet C; the closed end

hangs down into the firebox in contact with the hot furnace gases. The inner tube B is open at both ends and is suspended inside the outer tube by pins or feathers. The upper end of the inner tube is expanded so as to give a free entrance to the water.

The tubes being suspended in the firebox and exposed to intense heat, there is a rapid formation of steam, which rises to the surface through the space between the outer and inner tubes. A stream of water is thus kept continually flowing downwards through the inner tube to supply the place of the rising steam, and the result is a rapid and continual circulation within the tubes.

Field tubes are also used in other boilers. Sometimes they are hung in the furnace flues of Cornish and Lancashire boilers. They are also used in some forms of fire-engine boilers.

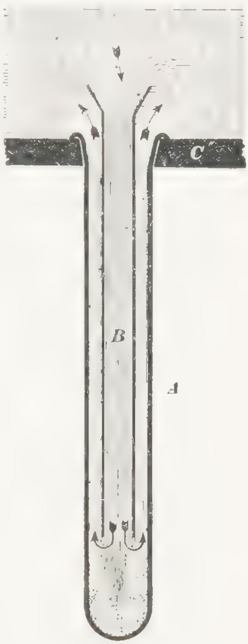


FIG. 25.

CIRCULATION OF WATER IN STEAM BOILERS.

35. Transmission of Heat.—The transfer of the heat generated by burning fuel in the furnace to the water in the boiler is accomplished, in the order named, by *radiation*, *conduction*, and *convection*.

36. Radiation of heat is the transfer of heat through space; **conduction** of heat may be defined as the transfer of heat through solids; and **convection,** as the transfer of heat through liquids or gases. The heat generated in the furnace is transmitted to the plates of the boiler principally by radiation. Convection also plays an important part in the

transfer of heat from the fire to the boiler; as the hot gases pass from the fire through the flues and tubes, currents are formed that bring successive portions into contact with the boiler surface, thus enabling them to give up their heat to the boiler. Heat is transmitted through the plates and tubes by conduction and through the water in the boiler by convection.

37. Circulation is the name given to the motion of the water under the influence of heat. The water nearest the plate becomes heated, expands, and thus becoming lighter, rises to the top. In consequence, cold water will flow in from the sides and take its place, become heated in turn, and rise. Now, the rapidity with which the transfer of heat by convection will take place naturally depends on the rapidity of the circulation. If this is interfered with, the transfer of heat will be slow; on the other hand, if the circulation is free, the transfer of heat will be rapid. To a small extent the transfer of heat depends on the quality and thickness of the material through which the heat passes by conduction; the influence of this, however, has experimentally been shown to be small.

38. Water circulation is essential to the efficient operation of a boiler, since with a good circulation more of the heat will be absorbed by the water and less will pass up the chimney. Furthermore, a rapid circulation to a certain extent prevents the deposit of sediment carried in with the feedwater. It also tends to keep all parts of the boiler at a uniform temperature.

39. The circulation of the water in a horizontal return-tubular boiler is shown in Fig. 26. The heated water rises from the hottest part of the bottom of the shell, which is directly above the furnace, thus carrying the steam bubbles to the surface. The cooler water from the rear of the boiler flows in to take the place of that which has risen, and thus circulation is maintained in the general direction indicated

by the arrows. It will be noticed that the horizontal direction of the circulation is contrary to that of the gases from the furnace. In cylinder boilers the water is contained in a solid mass, and there is nothing to interfere with the free circulation. In flue boilers and tubular boilers, however, the

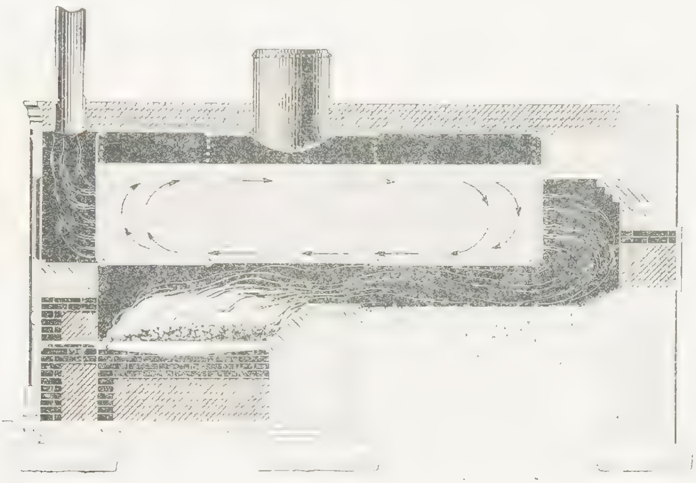


FIG. 26.

flues and tubes break up the body of water in such a manner as to cause numerous small currents that oppose the general direction of the circulation more or less. In order to have the circulation interfered with as little as possible, the tubes should not be spaced too closely together.

40. Uniform Circulation.—It is one of the strong points of correctly designed water-tube boilers that the circulation is strong and uninterrupted by any opposing currents. This is accomplished by passing the water always in the same direction through the series of tubes. The difference between the cylindrical and water-tube boilers in this respect may be illustrated as follows: The cylindrical boiler with its mass of water may be compared to an ordinary kettle in the process of boiling (see Fig. 27). The

water rises rapidly around the outer edges and flows downwards in the center. If, however, the fire is quickened, the upward and downward currents interfere with each other and the kettle boils over.

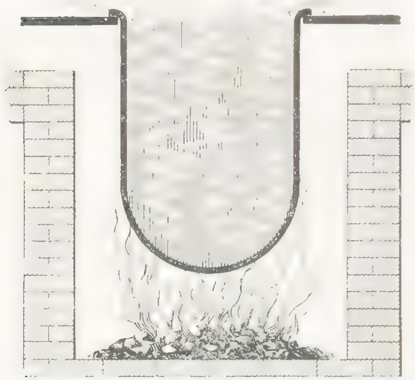


FIG. 27.

FIG. 28.

The water-tube boiler is and should be identical in principle, with a **U** tube depending from a vessel filled with water with the heat applied to one leg (see Fig. 28). The circulation is set up immediately and proceeds quietly no matter how fierce the fire may be.

BOILER DETAILS.

METHODS OF CONNECTING BOILER PLATES.

1. In the construction of a boiler, the following order of operations is usually followed in the shop: The flat plates from the rolling mill are cut or sheared to the desired size; after being sheared they are placed in a machine that planes the rough edges. Next the rivet holes are punched or drilled in the edge of the plate. The plate is then passed through large rolls and bent to a cylindrical form in such a manner that the corresponding rivet holes in the two edges come opposite one another. One or two bolts are put through to hold the edges together, and the plate is then riveted. The heads are flanged and riveted in place. After the riveting is completed, the tubes are put in and expanded, the stays are put in place, and the boiler is ready for its setting. The details ot construction will now be considered.

RIVETING.

2. Rivets.—Common forms of rivets are shown in Figs. 1 to 5. In Figs. 1 and 2 are shown examples of hand riveting; in the first case, the head is hammered down to a cone, while in Fig. 2 the rivet has a cup, or snap, head. This form of head is produced by first hammering the rivet down roughly and then finishing the head by a cup-shaped

die called the **button set.** The dotted lines in the figure

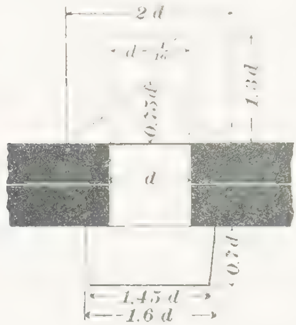


FIG. 1

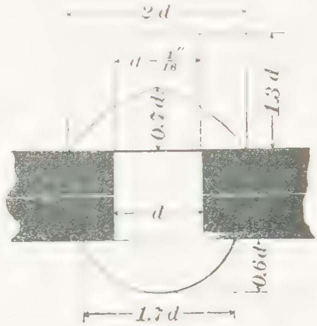


FIG. 2

show the shape of the rivet shank before being upset by the hammer.

3. The rivets shown in Figs. 3 and 4 are examples of machine riveting. The rivet is placed between two dies which are forced together by heavy steam or hydraulic pressure. The most important advantages of machine riveting

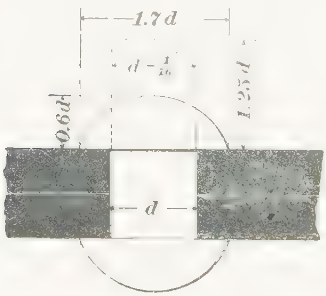


FIG. 3.

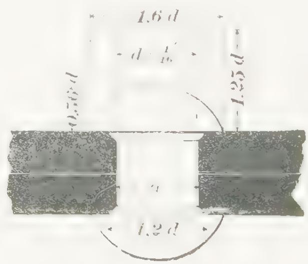


FIG. 4

are the following: By means of the force with which the plates can be held together while the head is being formed, a tighter joint can be made; the heavy pressure used to upset the rivet and form the head causes it to expand and fill the hole more completely than it will when headed by the blows of a hammer; when a large number of rivets is to be driven, machine riveting is cheaper than hand riveting.

A rivet with countersunk head is shown in Fig. 5. Such

riveting is sometimes necessary where a smooth surface is needed for the attachment of boiler mountings. In Figs. 4 and 5 the holes in the plates are countersunk slightly under the rivet heads. This provides for an increase in the size of the rivet just under the head and makes the rivet much stronger than is the case where the connection between head and rivet forms a sharp angle, as in Figs. 1, 2, and 3.

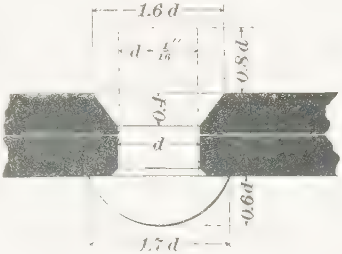


FIG. 5.

As shown in the figures, the rivets, before being headed, are slightly smaller than the hole, so that they may be inserted easily. It is the general rule to make the rivet hole $\frac{1}{16}$ inch larger than the rivet. When the work is properly done, the upsetting action of heading the rivets causes them to fill the holes when headed down.

4. Riveted joints of different forms are shown in Figs. 6 to 9. When one plate overlaps the other and the two are joined with one or more lines of rivets, as shown in Figs. 6 and 7, the joint is said to be **lap-riveted.** When, however, the plates are placed edge to edge, as in Figs. 8 and 9, and the joint is covered with one or two plates, the joint is called a **butt joint.**

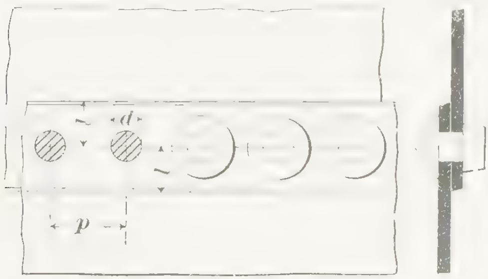
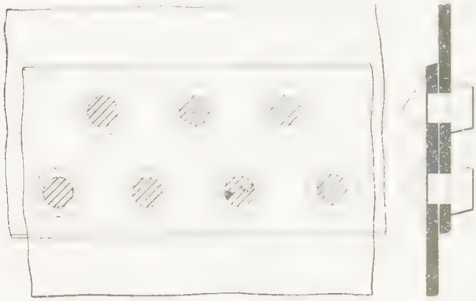


FIG. 6.

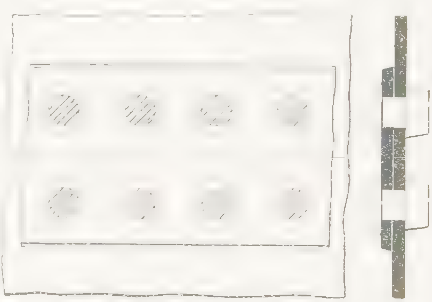
5. Fig. 6 represents a **single-riveted lap joint,** that is, the plates are overlapped and joined with one row of rivets. The distance p from center to center of the rivet holes is called the **pitch** of the rivets. The distance l from the center line of the rivet hole is usually made $1\frac{1}{2}$ times the diameter d of the rivet hole.

6. Fig. 7 shows a **double-riveted lap joint.** The rivets may be **staggered,** as shown in the figure, which method is commonly called **zigzag riveting,** or placed one behind the other, as in Fig. 9. In the latter case the joint is **chain-riveted.** In a zigzag riveted joint, the distance from the center of one rivet to the center of the next rivet in the other row is called the **diagonal pitch.**



It is quite customary in boiler construction to single-rivet the girth seams and double-rivet the longitudinal seams, since the stress on the latter is twice that on the former, as will be shown further on.

7. A **butt joint** with a single cover-plate is shown in Fig. 8, while a butt joint with two cover-plates is shown in Fig. 9. Either may be riveted with one, two, or more rows of rivets; Fig. 9 shows an example of chain-riveting. A well-designed butt joint with two plates will be stronger than one with a single plate. Butt joints are generally used for plates over



$\frac{1}{2}$ inch thick and are taking the place of lap joints in good designs of smaller work. When one cover-plate is used on a butt joint, its thickness should not be less than $1\frac{1}{4}$ times the thickness of the plate; when two cover-plates are used, the thickness of each should not be less than about $\frac{5}{8}$ of the plate thickness.

8. Arrangement of Joints and Plates.—The plates of externally fired boilers should be arranged so that the riveted

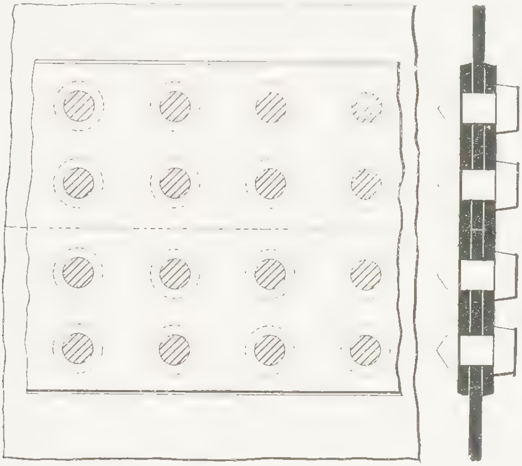


FIG. 9.

joints are as far as possible from the fire. This may be accomplished by using extra large plates for the furnace end of the shell.

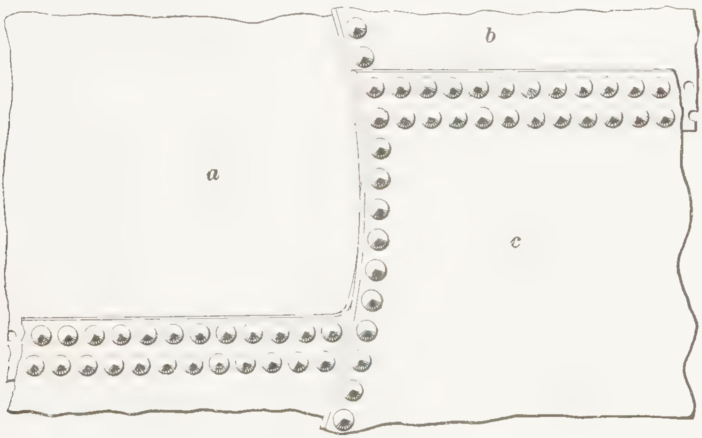
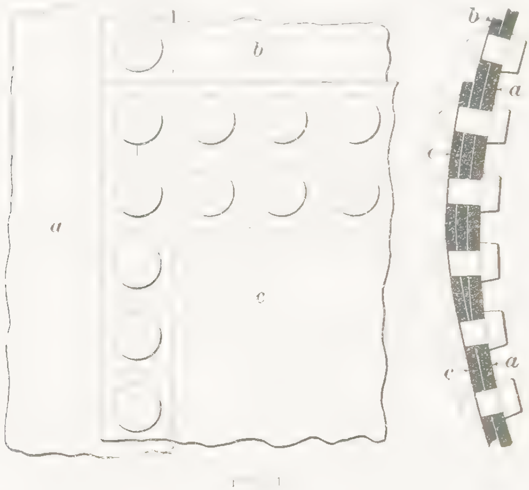


FIG. 10.

Wherever a girth seam occurs, the longitudinal seams should break joint, as shown in Fig. 10. In order to make

a tight joint where the three plates come together, the inner plate of the longitudinal joint must be hammered thin at the edge, as shown in Fig. 11.



In the construction of both vertical and horizontal shells, it is customary to have the inside lap facing *downwards*, since if it faces upwards, a ledge is formed on which sediment may be deposited.

Since wrought-iron plates are stronger in the direction of the fiber, they should be arranged so that the fiber runs circumferentially around the shell; that is, in the direction of the girth seams.

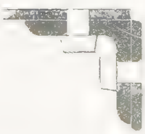


FIG. 12.

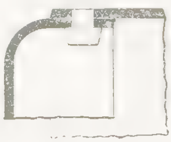


FIG. 13.

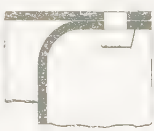


FIG. 14.

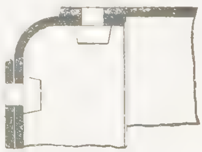


FIG. 15.

9. Connecting Plates.—Different methods of connecting plates at right angles are shown in Figs. 12 to 15. In

Fig. 12 the two plates are riveted to an angle iron. This construction is used sometimes for connecting the heads of a boiler to the shell. As shown in Figs. 13 and 14, the head is flanged and riveted to the shell, while in Fig. 15 the head and shell are connected by a flanged ring. The methods of connection shown in Figs. 13 and 14 are generally considered

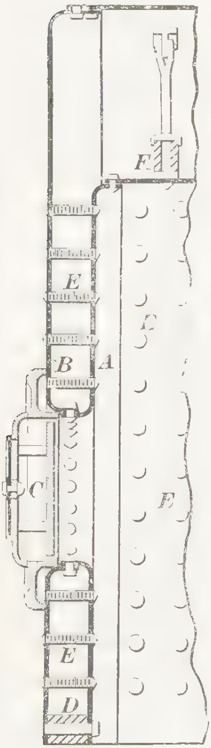


FIG. 16.

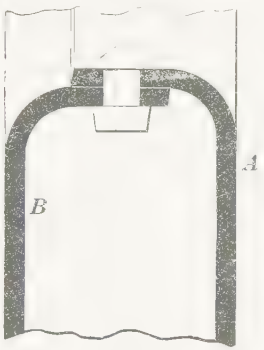


FIG. 17.

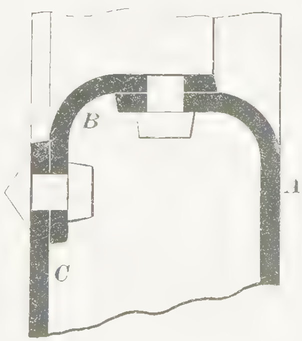


FIG. 19.

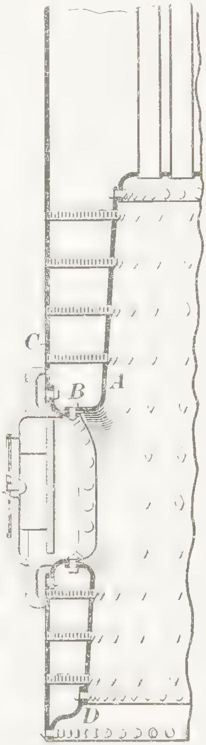


FIG. 18.

preferable to those shown in Figs. 12 and 15, since in the latter methods there are two joints to be kept tight, while in the former there is but one.

Iron or steel for flanging should be of the best quality. The radius of the curve to which the head is flanged should be at least 4 times the thickness of the plate.

Some makers of large boilers prefer to flange the end plates of the shell to receive the head, which is, consequently, a flat disk.

In Figs. 16 to 22 is shown the usual construction of the water legs and furnace doors of vertical and firebox boilers.

FIG. 20.

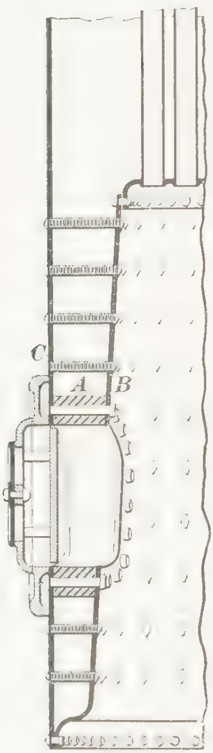


FIG. 21.

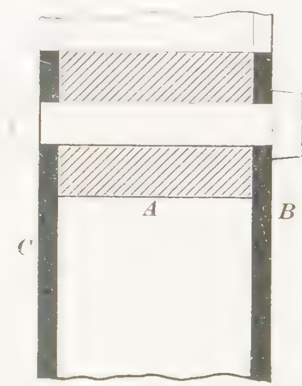


FIG. 22.

Fig. 16 shows the door constructed by flanging the furnace sheet A and the front sheet of the boiler B. In the figure the joint is single-riveted, although it is frequently double-riveted. An enlarged view of this construction is shown in

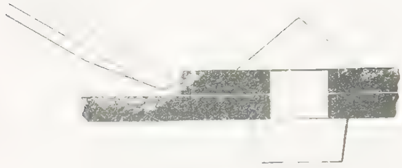
Fig. 17. The door *C* is generally made of cast iron and is hinged to a cast-iron frame that is usually held in position by four studs. Sometimes the frame is omitted and the door is made of wrought iron; it is then held in position by riveting the hinges to the boiler.

Around the lower ends of the water legs, or around the bottom of the furnace, and between the inside and outside plates is riveted a wrought-iron ring *D*. In cheap boilers this ring is frequently made of cast iron. Instead of flanging both sheets as in Figs. 16 and 17, the furnace opening is sometimes constructed as shown in Figs. 18 and 19. A hole is cut in the outer sheet *C*, and the furnace sheet *A* is flanged. The flanged ring *B* is then riveted to the plates *A* and *C*, and forms the opening for the door. An enlarged view of this construction is shown in Fig. 19. A flanged ring *D*, Fig. 18, is sometimes used at the bottom of the water leg in place of a wrought-iron ring *D*, Fig. 16, one of the flanges being riveted to the furnace plate and the other to the shell, as shown. An enlarged view of this construction is shown in Fig. 20. In Fig. 21 is shown another method of constructing the opening for the furnace door and the bottom of the water leg. In this construction the wrought-iron ring *A* is placed between the furnace plate *B* and the shell of the boiler *C*, and riveted to them. An enlarged view of this construction is shown in Fig. 22. At the bottom of the water leg the furnace plate is flanged and riveted to the shell, as shown.

10. Rivet holes are either punched or drilled. Punching, while cheaper than drilling, is generally believed to injure the plates, particularly if they are at all hard or brittle. Many makers punch the hole smaller than its intended diameter and then ream it out, thus cutting away the injured metal around the holes. Annealing the plates after punching will partly remove the injury. It is the present practice of good boilermakers to drill all plates, though this practice is not universal.

11. Calking is an upsetting process applied to a riveted joint in order to make it steam-tight. The operation is

shown in Fig. 23. A round-nose calking tool is driven against the beveled edge of the upper plate, forcing the metal in close contact with the lower plate and effectually closing the seam. A tool with a sharp edge should never be used, as it is liable to score the under plate and lead to grooving.



THE BOILER SHELL.

STRESSES ON BOILER SHELLS.

12. If the cylindrical shell shown in Fig. 24 is subjected to an internal pressure, there will be two forces tending to

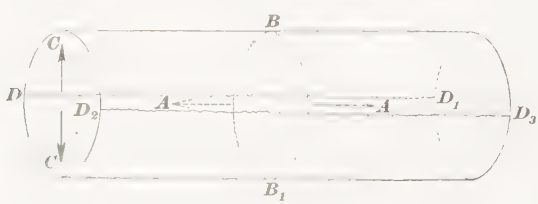


FIG. 24.

rupture it. One force, indicated by the arrows A, A, acting in the direction of the length, tends to tear the shell in a transverse plane, as $B\,B_1$. The other force, indicated by the arrows C, C, acting perpendicular to the axis, tends to rupture the boiler in a longitudinal plane passing through the axis, as $D\,D_1\,D_3\,D_2$. These two forces are opposed by the tenacity of the material of which the shell is composed. It is easy to see that the magnitude of the force tending to rupture the shell in a transverse plane is equal to the area of the head in square inches times the steam pressure per

square inch. As this force is resisted by the tenacity of the material, the magnitude of the tenacity being measured by the sectional area, the stress per square inch of section of the material is

$$\frac{\text{area of the head} \times \text{pressure}}{\text{area of section}}$$

13. It can be proved, both by mathematics and by experiment, that the magnitude of the force tending to rupture the shell in Fig. 24, in a longitudinal plane, is equal to the internal diameter multiplied by the length multiplied by the pressure.

To resist this force, we have the combined sectional area of the material of the two sides of the shell. Hence, the stress per square inch of section equals

$$\frac{\text{the internal diameter} \times \text{the length} \times \text{the pressure}}{\text{combined sectional area}}.$$

Suppose we have a plain cylindrical shell constructed of any convenient material. Let the inside diameter be 36 inches, the length 120 inches, the thickness of the shell $\frac{1}{4}$ inch, and the internal pressure to which it is subjected 100 pounds per square inch.

The pressure on the head and, consequently, the magnitude of the force acting in the direction of the length, is $36^2 \times .7854 \times 100 = 101,787.8$ pounds. This force is resisted by the tenacity of $36.5^2 \times .7854 - 36^2 \times .7854 = 28.471$ square inches of material. Hence, the stress per square inch of section is $101,787.8 \div 28.471 = 3,575.14$ pounds. The magnitude of the force acting perpendicular to the axis equals $36 \times 120 \times 100 = 432,000$ pounds. The area of material resisting this force equals $120 \times .25 \times 2 = 60$ square inches; hence, the unit stress $432,000 \div 60 = 7,200$ pounds per square inch. This shows that there is $7,200 \div 3,575.14$, or about twice as much resistance to transverse rupture as there is to rupture in a longitudinal plane. Hence, it follows that, if the material is proportioned to withstand the force perpendicular to the axis, it will possess ample strength in the other direction.

14. From the foregoing calculation, it is seen that the girth or transverse seams of a boiler need only be single-riveted when the longitudinal seams are double-riveted, or even triple-riveted. When the longitudinal seams are butt joints with two cover-plates and triple-riveted, the girth seams are usually double-riveted.

BURSTING AND SAFE WORKING PRESSURES.

15. Theoretical Bursting Pressure. For convenience in calculation, the length of the shell is taken as 1 inch. If a boiler is constructed of plates varying in thickness and tensile strength, the least thickness and the lowest tensile strength must be used in calculating the strength of the boiler.

In a cylinder that is on the point of bursting, the resistance of the material to rupture must be equal to the force tending to cause rupture. Hence, a cylinder is on the point of bursting if the product of the diameter and pressure equals the product of twice the thickness of the cylinder and the ultimate tensile strength of the material of which it is composed. It will be noticed that the length of the cylinder has not been taken into account; the length has been assumed to be 1 inch, for the reason previously given. From arithmetic it should be plain that the bursting pressure equals

$$\frac{\text{twice the thickness} \times \text{ultimate tensile strength}}{\text{diameter}}.$$

This may be simplified by using the radius of the cylinder instead of the diameter. Then, as the radius is one-half the diameter, the bursting pressure would be

$$\frac{\text{thickness} \times \text{ultimate tensile strength}}{\text{radius}}.$$

Let t = thickness of cylinder in inches;
R = internal radius in inches;
S = ultimate tensile strength of the material;
P = bursting pressure in pounds;
p = safe working pressure in pounds.

Rule 1.—*To find the bursting pressure of a cylinder, divide the product of the thickness of the cylinder and the ultimate tensile strength of the material by the internal radius.*

Or,
$$P = \frac{t\,S}{R}.$$

Example.—A cast-iron pipe is 10 inches diameter and $\frac{1}{2}$ inch thick. The tensile strength of the iron is 12,000 pounds. At what pressure will the pipe burst?

Solution.—Applying rule **1**, we get

$$P = \frac{\frac{1}{2} \times 12{,}000}{\frac{10}{2}} = 1{,}200 \text{ lb. per sq. in.} \quad \text{Ans.}$$

16. The **efficiency of a joint** may be defined as the ratio of the strength of the joint to that of the solid plate. It is usually expressed in per cent., the strength of the solid plate being considered as the unit (100). Thus, if the efficiency of a joint is 56 per cent., it means that the strength of the joint bears the same proportion to the strength of the solid plate that 56 does to 100.

17. The **average efficiencies of riveted joints** are: for a single-riveted lap joint, 56 per cent.; for a double-riveted lap joint, 70 per cent.; for a triple-riveted lap joint, 75 per cent.; for a double-riveted butt joint with two cover-plates, 76 per cent.; for a triple-riveted butt joint with two cover-plates, 85 per cent.; single-riveted and double-riveted butt joints with one cover-plate give about the same efficiencies as single-riveted and double-riveted lap joints.

18. Rule **1** applies to a cylinder *without* a seam weaker than the material of which the cylinder is constructed. But the longitudinal joints of the shell of a boiler are considerably weaker than the solid plate, and, consequently, a boiler shell will rupture at a pressure depending on the strength of its weakest part. Let f denote the efficiency of the joint, and let the other letters have the same meaning as in the previous rule. Then, the bursting pressure of a boiler shell may be obtained from the following rule:

Rule 2.—*Multiply the thickness of the material by its tensile strength and by the efficiency of the joint, in per cent. Divide the product by 100 times the internal radius.*

Or, $$P = \frac{t\,S\,f}{100\,R}.$$

In engineers' examinations, the calculation of the bursting pressure is one of the questions usually asked of candidates. Sometimes the efficiency of the joint is given, and sometimes the candidate is merely told what kind of joint is used. In the latter case, use the average efficiency corresponding to the kind of joint, as given in Art. **17.**

EXAMPLE.—A boiler 48 inches in diameter is constructed of steel plate $\frac{1}{4}$ inch thick, having a tensile strength of 55,000 pounds per square inch. The longitudinal seam being a double-riveted lap joint, what is the bursting pressure?

SOLUTION.—By Art. **18,** the average efficiency of a double-riveted lap joint is 70 per cent. Then, applying rule **2,** we get

$$P = \frac{\frac{1}{4} \times 55{,}000 \times 70}{100 \times \frac{48}{2}} = 401 \text{ lb. per sq. in., nearly. Ans.}$$

19. Safe Working Pressure.—All authorities are agreed that a boiler should not be worked at a pressure near the bursting pressure, but they disagree considerably upon what ratio the safe working pressure should bear to the bursting pressure. This will account for the different results obtained by using different formulas. It is believed that it is the most general practice to make the ratio of the safe working pressure to the bursting pressure 1 : 5. Using this ratio, we have the following rule for safe working pressure:

Rule 3.—*Multiply the thickness of the material by its ultimate tensile strength and by the efficiency of the joint, in per cent. Divide the product by 500 times the internal radius.*

Or, $$p = \frac{t\,S\,f}{500\,R}.$$

EXAMPLE.—Taking the previous example, what working pressure would you allow?

SOLUTION.—Applying the rule just given, we get

$$p = \frac{\frac{1}{4} \times 55,000 \times 70}{500 \times \frac{48}{2}} = 80.2 \text{ lb. per sq. in., nearly. Ans.}$$

20. Board of Supervising Inspectors' Rules.—The Board of Supervising Inspectors of Steam Vessels prescribe the following rules, which, while only in force for marine boilers, are frequently used for stationary boilers:

Rule 4.—*To find the safe working pressure of a cylindrical boiler having single-riveted longitudinal seams, divide the product of the thickness of the shell and the ultimate tensile strength of the material by 6 times the radius.*

Or,
$$p = \frac{t\,S}{6\,R}$$

EXAMPLE.—What safe working pressure would be allowed on a boiler with single-riveted longitudinal seams constructed of material $\frac{3}{8}$ inch thick and having a tensile strength of 50,000 pounds per square inch. The boiler is 48 inches in diameter.

SOLUTION.—Applying rule 4, we get

$$p = \frac{\frac{3}{8} \times 50,000}{6 \times \frac{48}{2}} = 130.2 \text{ lb. per sq. in. Ans.}$$

Rule 5.—*To find the safe working pressure of a cylindrical boiler having double-riveted longitudinal seams, divide the product of the thickness of the shell and the ultimate tensile strength of the material by 5 times the radius.*

Or,
$$p = \frac{t\,S}{5\,R}.$$

EXAMPLE.—A cylindrical boiler has the following dimensions: Diameter, 36 inches; thickness of shell, .25 inch; ultimate tensile strength of the material, 45,000 pounds per square inch. The seams being double-riveted, find the safe working pressure.

SOLUTION.—Applying rule 5, we have

$$p = \frac{.25 \times 45,000}{5 \times 18} = 125 \text{ lb. per sq. in. Ans.}$$

Rules 4 and 5 are based on a ratio of bursting pressure to safe working pressure of about 1 : 3.5; a ratio rather smaller than usually deemed advisable in stationary work.

21. The rules here given are applicable to new boilers. While in use, the plates will gradually waste away or lose in strength by the phenomena called grooving and honeycombing. In that case, the least thickness of plate should be used in calculating the working pressure.

Furthermore, the student is cautioned against assuming that the strength of the shell by itself determines the safe working pressure. A steam boiler is like a chain in that its weakest part determines its strength. The strength of the bracing will also have to be taken into account, and if the bracing is weaker than the shell, the pressure must be reduced to suit the bracing.

STAYING OF FLAT SURFACES.

22. Surfaces That Require Staying.—The surfaces of steam-boiler shells are, in general, either cylindrical, hemispherical, or flat. A cylinder or a sphere subjected to an internal pressure is *self-supporting;* that is, the pressure tends to maintain the cylindrical or spherical form of the vessel and hinders distortion instead of producing it. If the vessel is composed of flat surfaces, however, an internal pressure tends to distort it and give it an approximately spherical form. Then, as flat surfaces are not self-supporting, it follows that they must be stayed or braced.

23. The flat surfaces commonly found in stationary boilers are the boiler heads of the ordinary return-tubular boiler, and the heads, flat sides, and tops of fireboxes of internally fired boilers of the locomotive type.

DIAGONAL STAYS.

24. Crowfoot Braces.—The heads of stationary boilers are supported in various ways. Probably the most common method is to support them by **crowfoot braces,** which are securely riveted to the head and the shell. The crowfoot brace is shown in Fig. 25 (*a*). In this style of brace the

crowfoot, or part that is riveted to the head, is formed by welding flat bars to a cylindrical stem. The strap end, or part that is riveted to the shell, is also welded to the stem. An improved form of crowfoot brace is the **McGregor brace**, shown in Fig. 25 (*b*). This brace is formed by a piece of sheet steel, bent in one heat as shown. Being weldless, it may naturally be assumed, and the assumption has been borne out by experiments, that, for equal cross-sectional areas, it will bear a much greater strain than the welded crowfoot brace.

FIG. 25.

It will be observed that the crowfoot of the McGregor brace is formed by splitting the sheet and bending it at a right angle. In the **Huston** improved crowfoot brace, shown in Fig. 25 (*c*), the crowfoot is formed by flanging the plate of which the brace is formed, thus giving probably the strongest form of crowfoot that can be devised.

25. Another form of brace used occasionally, and especially for staying the lower part of the head of return-tubular boilers below the nest of tubes, is shown in Fig. 26. The end *A*, instead of being riveted to the head, is threaded and supplied with nuts and taper washers on each side of the head, as shown in Fig. 26. The washers have such a taper that when one of the faces rests against the head, the

FIG. 26.

other is parallel to the faces of the nuts. The hole through which the stay passes is not threaded, but is made sufficiently large to allow the stay to pass through. In cheaper work, the brace is bent near the head, in order to pass through the head at a right angle, thus doing away with the taper washers. Mechanically considered, this is not such a good arrangement as the one shown in Fig. 26, since the stress on the stay will tend to straighten out the bend and thus allow the head to buckle and give more than with the other design.

26. Gusset stays are occasionally used for staying the heads in large boilers. One of these stays is shown in Fig. 27. A gusset stay consists of an iron or steel plate, as *A*, which is placed between angle irons, as *B*, *B*, which are securely riveted to the head and shell. The plate in turn is riveted to the angle irons. Gusset stays interfere considerably with examination and repair of the boilers, and owing to their absence of flexibility, often induce grooving and cracking of the head-plate. For these reasons they have not found much favor in the United States; however, they are largely used in Europe.

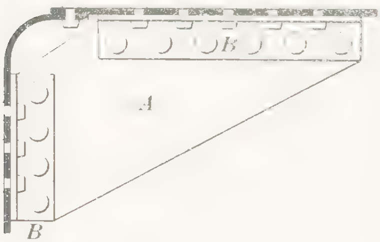


Fig. 27.

27. Bracing With Angle Irons or T Irons.—Small boilers carrying a rather low steam pressure occasionally have the heads braced by angle irons or T irons riveted to the head. For higher steam pressures, the angle irons or T irons may in turn be supported by braces, as shown in Fig. 28, one end of which is riveted to the shell and the other end either forked to straddle the leg of the T iron or flattened to go between two angle irons. Connection between the angle iron or T iron and the brace is then made

by a cotter, and, in better work, by a bolt and nut. The stays so far shown are known technically as **diagonal** stays, so called because they are inclined to the surfaces they support.

DIRECT STAYS.

28. For large boilers carrying high pressures, the most direct way of supporting the heads is to use through stayrods spaced about 14 inches from center to center. These, in the best construction, as shown in Fig. 29, are rods passing through both heads and provided with nuts and washers on both sides of the heads. A large washer on the outside, or a reenforcing leaf, is often riveted to the head, which reduces the bulging under pressure and allows the stayrods to be widely spaced. Channel irons riveted to the inside of the head, and in turn supported by stayrods, have also been used for high-pressure boilers.

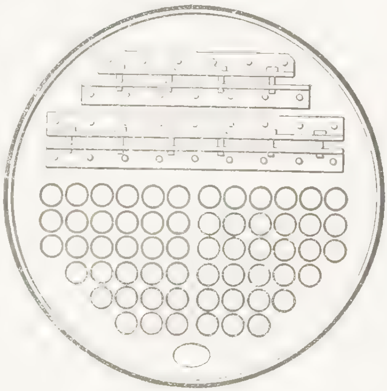


FIG. 28.

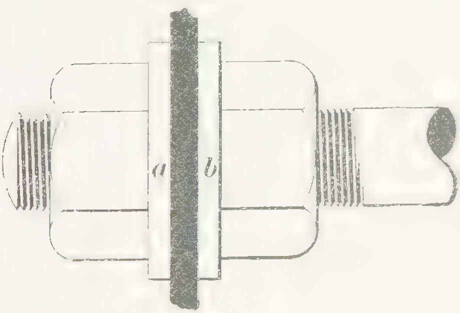


FIG. 29.

29. Staybolts.—For staying the flat surfaces of the water legs of boilers of the locomotive type, the usual and most direct method is to use screw staybolts, one of which is shown in Fig. 30. These screw staybolts are screwed

both into the inside and outside sheets and then riveted over at both ends while cold. Staybolts for these surfaces being subjected to a great deal of alternating bending stress

Fig. 30.

(in addition to a direct tensile stress) will sooner or later break off close to the sheets. In order to give warning of this, in good work, holes as *a* in Fig. 31 are

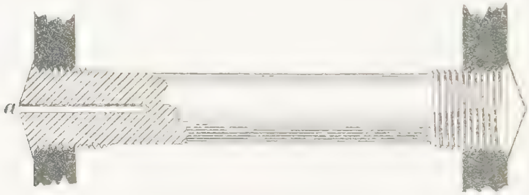


Fig. 31.

drilled axially in the center of each staybolt and extend at least ½ inch beyond the internal surface of the sheets. Should a bolt break, the steam or water issuing from the hole will give warning of the break. In boilers coming under the supervision of the United States Steam Vessel Inspection Service, screw staybolts are not allowed to be used any more unless they are drilled at each end in the manner explained.

30. Hollow Forged Staybolts.—Within the last few years hollow forged staybolts have been placed on the market, in which a hole extends axially all through the bolt. Staybolts of this kind possess more flexibility than solid bolts, and, of course, will give due warning of a break no matter where it may occur.

Instead of riveting over the end of staybolts, they may be fitted with a nut on the outside of the sheet. While this is

the common practice in marine boilers of the Scotch type, it is rarely done in stationary boilers.

For high-grade work some boilermakers will turn down the thread on the part of the staybolt between the two sheets, as shown in Fig. 31. This is an excellent practice, as it not only renders the staybolt more flexible, but also tends to reduce corrosion by exposing a much smaller surface to the corrosive action of the water. The stays here shown are at a right angle to the surface they support, and are called **direct stays.**

GIRDER STAYS.

31. The upper plate or **crown sheet** of the furnace of internally fired boilers of the locomotive type is supported

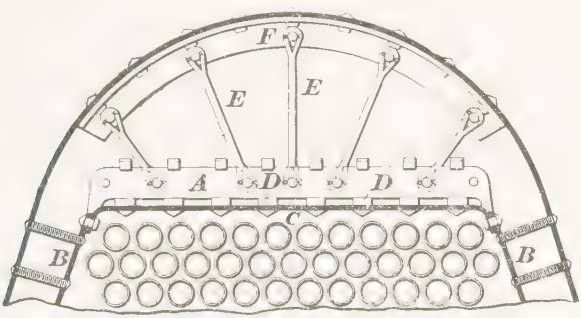


FIG. 32.

in various ways. Probably the most common method is to support it by staybolts screwed and riveted to the sheet and suspended from **girder stays,** as *A* in Fig. 32. These girder stays, one of which is shown in detail in Fig. 33, may in turn be supported by the **sling stays** *E*, *E*, Fig. 32, which are cottered to the angle-iron ring *F*, or may instead be riveted directly to the shell. Some makers of locomotive-type boilers condemn this arrangement as interfering too much with examination and repair;

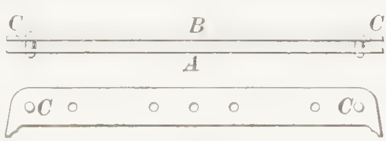


FIG. 33.

they make the girder stays of sufficient strength to support the staybolts. Two methods of suspending the staybolts are shown in Fig. 34, of which the one shown at the left is the most common one. The head is formed with lugs *D*, *D*, which prevent the spreading of the girders. When the crown sheet of a locomotive-type boiler is cylindrical, or approximately cylindrical, it is often supported by long stays screwed and riveted both into the shell and into the crown sheet.

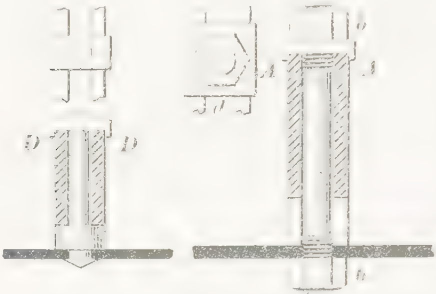


Fig. 34.

32. Fastening Tubes to Heads.—Boiler tubes up to 5 inches in diameter are generally fastened and made steam-tight by expanding them into holes cut into the heads. The tubes are cut off to a length about $\frac{1}{2}$ inch in excess of the outside length between the sheets which are to receive them; they are then driven into place until half of their excess in length projects beyond each tube sheet. The expander is now inserted and the tube expanded tightly into its hole; the projecting ends of the tubes are then nicely beaded over. Tubes and flues above 5 inches in diameter are commonly riveted to the heads, which are flanged to receive the tube.

33. For expanding the tubes, a **Dudgeon roller-tube expander,** shown in Fig. 35, is most commonly used. It

Fig. 35.

consists of a body A provided with three slots for the reception of the rollers a, a, a. A plate B, fastened to the body by the three screws c, c, c, prevents any longitudinal movement of the rollers. The rollers are forced outwards and rotated by the taper pin C, which is provided with a head perforated with two holes at right angles to each other. To make the expander adjustable for different thicknesses of tube sheets, the hood D may be moved longitudinally and may be locked in any desired position by means of a small taper pin d, which is driven inwards to lock the hood. The operation of the expander is as follows: The expander is pushed into the tube, the projections D', D'' limiting the depth to which the tool may be inserted. A sharp blow with a copper hammer is struck on the head of the pin C, forcing it inwards, and, hence, the rollers outwards. Next, a bar is inserted into one of the holes in the head and the pin rotated; the friction between the pin and the rollers causes the rollers, and, consequently, the whole tool, to rotate; this operation expands the tube.

BOILER FITTINGS.

THE SAFETY VALVE.

1. Purpose of Safety Valve.—The safety valve is attached to the boiler to prevent the steam pressure from rising above a certain point. Suppose a boiler is generating steam faster than the engine uses it; it is plain that a large quantity of steam is crowded continually into the steam space of the boiler. The necessary result is a rise in the pressure of the steam. If the pressure continues to rise, it soon becomes greater than the boiler is able to withstand, and an explosion occurs. It is the duty of the safety valve to prevent this increase of pressure.

2. Weighting of Safety Valves.—The safety valve consists simply of a plate, or disk, fitting over a hole in the boiler shell. This plate is held to its place in one of three ways: (1) By a dead weight; (2) by a weight on a lever; (3) by a spring.

The weight or spring is so adjusted that when the steam reaches the desired pressure, the disk is raised from its seat, and the surplus steam escapes through the opening in the shell.

3. Dead-Weight Safety Valve.—The dead-weight form of safety valve, while quite popular in Europe, has not found favor in America, since the high steam pressures now used require an extremely heavy weight. Dead-weight safety valves possess one great advantage over other types: it is difficult to overload them. The weights are so large that it would take a considerable added weight to materially raise the blowing-off pressure, and any added weight could readily be seen by the engineer.

4. Lever Safety Valves.—Two forms of lever safety valves are shown in Figs. 1 and 2. The valve *V* is held to

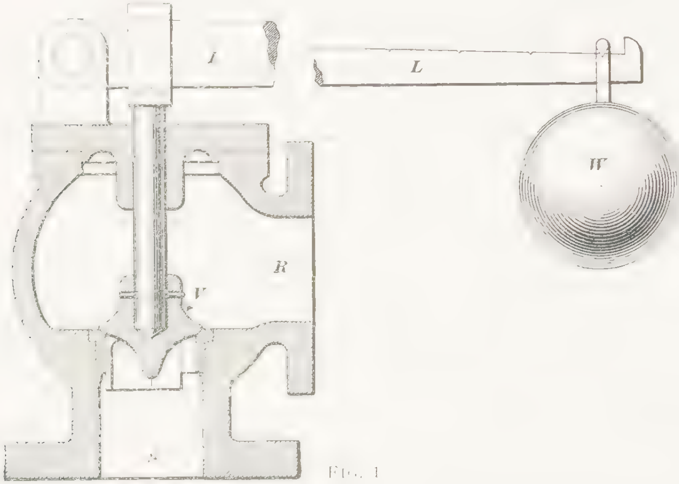


FIG. 1

its seat by the weighted lever *L*. The weight *W* is adjustable along the lever, so that the valve may be set to blow off at various steam pressures.

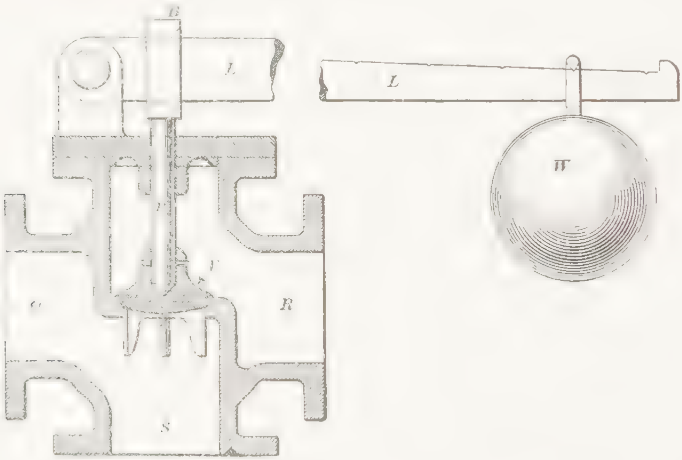


FIG. 2.

The valve shown in Fig. 1 is attached directly to the boiler shell; the steam enters from the boiler at *S* and is

discharged through the orifice *R*. The valve shown in Fig. 2 differs from the other in being attached to the supply pipe. The steam passes on its way from the boiler through the passage *S O*. When the pressure rises above the normal pressure, the valve *V* opens and the steam escapes into the air through the opening *R*.

The lever *L* is usually provided with several notches, each notch corresponding to a certain blow-off pressure. It is plain that the nearer the end of the lever the weight is placed, the higher will be the blow-off pressure.

5. Spring safety valves are largely used at present, especially on locomotive and marine boilers. The valve is held to its seat by a spring acting either directly on it or on a short lever. The Crosby "pop" safety valve is shown in Fig. 3. The main valve *V* is held down on the two circular seats *M* and *N* by the spring *S* acting on the rod *T*. The outer seat *N* is formed on the body *A* of the valve, while the inner and smaller seat *M* is formed on the upper edge of a cylindrical chamber *B* that is connected to the body *A* by arms containing the passages *C*, *C*. The hollow chamber *B* forms a guide for the valve *V*. Ordinarily, the steam exerts a pressure on the space between *M* and *N*; when the valve rises a little, the

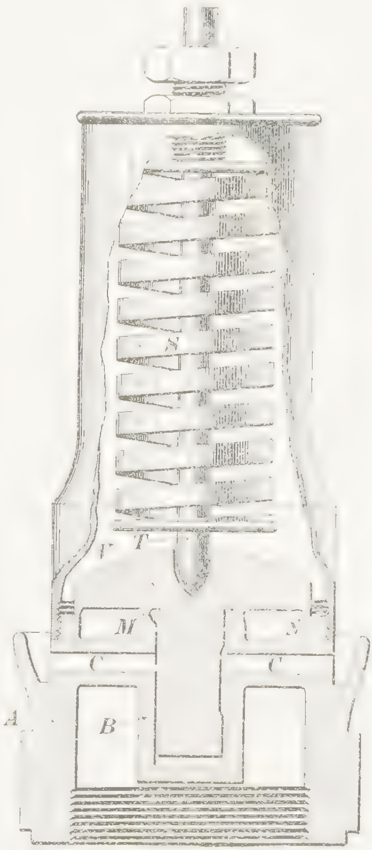


FIG. 3.

steam rushes over the seat N into the air and over the seat M into the chamber B, whence it escapes through the channels C, C. The channels are, however, not large enough to allow the steam to escape from the chamber as fast as it enters, and hence the pressure in the chamber rises and acts on the area inside the seat M. This additional pressure throws the valve wide open and quickly relieves the pressure in the boiler.

In this form of valve the blow-off pressure may be changed by altering the tension of the spring; thus, screwing down the hollow bolt at the top of the spring (Fig. 3) compresses the spring and raises the blow-off pressure.

DIRECTIONS FOR USE AND CARE OF SAFETY VALVES.

6. See that the safety valve is attached directly to the boiler. If there is a stop-valve between the valve and boiler, have it removed or arranged so that it cannot be shut.

Take care that the valve does not become corroded and stick to its seat. It is a good plan to frequently lift the valve from the seat and see whether or not it works freely.

Do not overload the valve or increase the tension of the spring, and take care that it is not done by others.

When the blow-off pressure is fixed, the weights are often locked in position by the boiler inspector and should not be changed.

7. In practice, the position of the weight on the lever is usually found by trial in preference to finding it by calculation. Most safety-valve levers are notched and have figures stamped below the notch, which are supposed to represent the pressure per square inch at which the valve will blow off when the weight rests in the notch. However, since it may be possible that the notches have not been correctly located, it is good practice to check the graduation by an actual trial. To do so, get up steam on the boiler, and as

soon as the steam gauge shows the blow-off pressure, shift the weight until the valve just commences to blow off. Then fasten or lock the weight, if possible, so it may not be shifted accidentally. Before adjusting the position of the weight, make sure that all parts of the valve work freely and that the steam gauge is correct.

8. After adjustment, the valve should occasionally be tested by comparing its blowing-off point with the pressure shown by the steam gauge. If the steam gauge indicates a higher pressure, it shows one of two things: either the steam gauge has become impaired or the valve is out of order. If there is reason to suspect the steam gauge, have it tested. At any rate, however, in order to be on the safe side, the steam gauge may be assumed to be correct and the valve then examined to see if everything works freely. If found so, and the weight is still at the same mark, it is reasonable to conclude that the gauge is out of order.

9. It is common practice in some localities to connect a pipe to the blow-off side of the safety valve for the purpose of carrying the steam blown off out of the boiler room. Such an escape pipe, while harmless enough when of sufficient area and kept well drained, may become a source of danger if no provision is made for draining it constantly. Instances are not rare where, owing to the absence of a drain pipe, the escape pipe has become filled with water, thus adding greatly to the external force on the valve and rendering it inoperative for the blow-off pressure for which it was set. When an escape pipe is used at all, it should not be of smaller diameter than the valve, and should have a drain pipe of ample size at its lowest point. No cock or valve should under any circumstances be placed in this drain pipe. Many engineers will not allow an escape pipe to be used under any consideration, claiming that with it, it is frequently impossible to know by sound whether the safety valve is blowing off. Safety valves should in all cases be so fitted that it is an *absolute* impossibility to shut off connection between the boiler and its safety valve.

CALCULATIONS RELATING TO SAFETY VALVES.

10. General Principles.—No safety valve can open without a slight increase of pressure above that for which it is set, since, in order to lift the valve, the pressure on the under side of the valve, which we may call the internal or upward force, must exceed the external or downward force on the valve plus the friction of the mechanism of the valve. If the internal and the external forces upon the valve are equal, the valve will be in equilibrium (balanced), and an increase of the internal force will cause it to open. A safety valve will not close until the pressure has been reduced somewhat below the pressure at which the valve opened.

The point at which a safety valve will blow off depends on the external force on the valve. To be in equilibrium, the external load exerting a downward pressure on the valve must be equal to the internal force exerting an upward pressure on the under face of the valve. Evidently the upward pressure is equal to the area of the valve multiplied by the pressure per unit of area.

Whenever the word "pressure" is used in relation to calculations pertaining to safety valves, the **gauge pressure** is meant, unless otherwise stated.

NOTE.—The area of a safety valve is that part that receives the steam pressure when the valve is seated.

Suppose that we have a dead-weight safety valve having a diameter of 4 inches and an external load or force consisting of the valve and stem, the supporting plate, and the weights, equal to 815.8 pounds; it is desired to know the pressure at which the valve will open. Since the internal and external forces must balance, it is evident that $815.8 = 4^2 \times .7854 \times$ steam pressure in pounds per square inch.

From this we get $\frac{815.8}{4^2 \times .7854} = 65$ pounds per square inch, nearly, the pressure at which the valve is about to open.

11. In the lever safety valve shown in Fig. 2, the external load depends on the position of the weight W on the lever L.

Here the same general law holds good; the external and the internal forces must be equal before the valve is about to open. The internal force, as stated before, is the area of the valve times the steam pressure. The downward force on the valve may be found as follows: Suppose we place the weight P, Fig. 4, weighing 100 pounds, directly

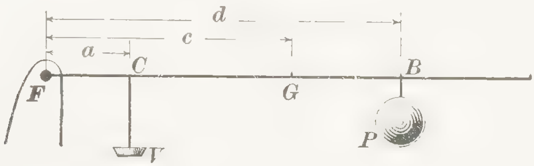


FIG. 4.

on top of the valve stem C. Evidently the downward force is now equal to the weight of the weight P. Suppose now that the weight is removed to the position shown in the figure, the weight's distance d from the fulcrum being 6 times greater than the distance a from the fulcrum to the center line of the valve. Evidently the effect of the weight on the valve stem will now be 6 times greater; that is, the downward force will be $6 \times 100 = 600$ pounds. Hence, to find the downward force, we divide the distance d by the distance a and multiply the quotient by the weight of the weight P.

As the valve and stem have a certain weight, the external force is increased an amount equal to that of the weight of the valve and stem in pounds. Furthermore, the lever has a certain weight, and this, acting at the center of gravity of the lever, adds a certain amount to the downward force. This amount is equal to the product of the distance from the fulcrum of the lever to its center of gravity and the weight of the lever, divided by the distance from the fulcrum to the center line of the valve.

12. The distance to the center of gravity of the lever may be found by balancing the lever on a knife edge and

measuring the distance from the center of the fulcrum to the knife edge. If this should not be feasible, the center of gravity must be found by calculation. In engineers' examinations, the lever is usually given as straight and parallel, in which case the distance from the fulcrum to the center of gravity of the lever should be taken as equal to one-half the length of the lever.

The amount of the downward force on the valve due to the weight of the lever may be found directly by attaching a spring balance by a cord to the lever at the point at which it acts upon the valve stem. The spring balance will indicate the correct downward force in pounds.

Now, to have the valve balance, the area of the valve times the steam pressure (the upward force) must equal the weight times the distance from the fulcrum to the weight divided by the distance from the fulcrum to the center line of the valve; to this downward force must be added the additional downward force due to the weight of the valve, stem, and lever (the external force).

13. Illustrative Example.—How to find the pressure per square inch at which a safety valve is about to blow off may best be explained by the following example: Suppose that we have a safety valve of the following dimensions: Let the area of the valve be 12.566 square inches; the distance from the fulcrum to the center line of the valve, 4 inches; the weight is to weigh 135.2 pounds; the length of the lever is to be 36 inches; the weight of the valve and stem, 9.2 pounds; and the downward force due to the weight of the lever, as found by one of the three methods previously explained, 150 pounds. From what has been explained above, it should be clear that the valve balances, or is in equilibrium, if the steam pressure $\times$ 12.566 $= 135.2 \times 36 \div 4 + 9.2 + 150$. That is, the steam pressure $\times$ 12.566 $= 1,376$.

From arithmetic, it should be plain that the steam pressure $= \frac{1,376}{12.566} = 109.5$ pounds per square inch.

14. Rules for Safety-Valve Calculations.—

Let A = area of valve in square inches;
D = distance from center line of valve to fulcrum, measured in inches;
L = distance of weight from fulcrum in inches;
P = steam pressure in pounds per square inch;
W = weight of the weight on lever in pounds;
w = weight of valve and stem in pounds plus the downward pressure due to weight of lever.

Rule 1.—*To find the pressure at which a safety valve is about to blow off, multiply the weight by the length of the lever and divide this product by the distance from the fulcrum to the center line of the valve. To the quotient add the downward pressure on the valve due to the weight of the valve, stem, and lever, and divide the sum by the area of the valve.*

Or, $$P = \frac{\frac{W L}{D} + w}{A}.$$

EXAMPLE.—The area of a lever safety valve is 11 square inches; the distance from the center line of the valve to the fulcrum, 4½ inches; the distance of the weight from the fulcrum, 35 inches; its weight, 125 pounds; the weight of valve and stem plus the downward pressure due to the weight of the lever equals 137 pounds. Find the pressure per square inch at which the valve is about to open.

SOLUTION.—Applying rule **1**, we have

$$P = \frac{\frac{125 \times 35}{4.5} + 137}{11} = 100.3 \text{ lb. per sq. in.} \quad \text{Ans.}$$

To explain how to find where a given weight must be placed on the lever in order that the safety valve may be about to blow off at a given pressure, we will take the example previously made use of. The pressure was found to be 109.5 pounds per square inch; hence, the total upward force is $109.5 \times 12.566 = 1{,}375.977$, say 1,376 pounds. This force is partially balanced by the weight of the valve and stem and the downward force due to the weight of the lever. Consequently, the total upward force $= 1{,}376 - (150 + 9.2) = 1{,}216.8$ pounds, is to be balanced by the downward force.

As the downward force, as previously explained, is the weight times the length of the lever divided by the distance from the fulcrum to the center line of the valve, it should be plain that the valve is in equilibrium again, if $1,216.8 = \frac{135.2 \times \text{the lever}}{4}$. That is, $1,216.8 = 33.8 \times$ the lever.

Hence, length of lever $= \frac{1,216.8}{33.8} = 36$ inches.

Rule 2.—*To find the distance from the fulcrum to where the weight must act in order to have the valve blow off at a given pressure, subtract the downward force due to the weight of the valve, stem, and lever, from the product of the area and the steam pressure. Multiply the remainder by the distance from the fulcrum to the center line of the valve and divide this product by the weight.*

Or,
$$L = \frac{(A P - w) D}{W}.$$

EXAMPLE.—At what distance from the fulcrum must a weight of 150 pounds act in order that the valve may be about to blow off at 100 pounds pressure, the diameter of the valve being $3\frac{3}{4}$ inches; the distance from the fulcrum to the center line of the valve $4\frac{1}{2}$ inches; and the downward force due to the weight of valve, stem, and lever 125 pounds?

SOLUTION.—Applying rule **2**, we get, since area $= (3\frac{3}{4})^2 \times .7854 = 11.04$ square inches,

$$L = \frac{(11.04 \times 100 - 125) \times 4.5}{150} = 29.37 \text{ in. Ans.}$$

15. It is desired to find the weight that must be placed on a lever to have the valve blow off at a given pressure:

Using the same example as in the other explanations, the unbalanced upward force, as previously found, is 1,216.8 pounds.

The valve balances if $1,216.8 = \frac{\text{weight} \times 36}{4}$.

That is, $1,216.8 = \text{weight} \times 9$, and the weight $= \frac{1,216.8}{9} = 135.2$ pounds.

Rule 3.—*To find the weight that must act on a lever at a given distance from the fulcrum so that the valve is about to blow off at a given pressure, subtract the downward*

force due to the weight of the valve, stem, and lever from the product of the area and the steam pressure. Multiply the remainder by the distance from the fulcrum to the center line of the valve, and divide this product by the distance from the fulcrum at which the weight is to act.

Or, $$W = \frac{(A P - w) D}{L}.$$

EXAMPLE.—A safety valve has the following dimensions: Area of the valve, 15.7 square inches; length of lever, 48 inches; distance from fulcrum to center line of the valve, 5 inches; the downward force due to the weight of the valve, stem, and lever is 182 pounds. Find the weight if the valve is about to blow off at 64 pounds pressure.

SOLUTION.—Applying rule **3**, we get

$$W = \frac{(15.7 \times 64 - 182) \times 5}{48} = 85.71 \text{ lb. Ans.}$$

NOTE.—The student should thoroughly familiarize himself with the calculations pertaining to lever safety valves, as a candidate for an engineer's license is usually asked to solve safety-valve problems similar to those given here.

Spring-loaded and pop safety valves are adjusted under pressure by comparison with an accurate steam gauge. No rules can be given by which to calculate the point at which they will blow off.

EXAMPLES FOR PRACTICE.

1. A dead-weight safety valve having an area of 12 square inches is to be on the point of blowing off at 75 pounds pressure, absolute; find the weight. Ans. 723.6 lb.

2. At what pressure will a safety valve of the following dimensions blow off: Area of valve, 10 square inches; distance from the valve to the fulcrum, 3 inches; length of lever (the distance from the fulcrum to the point where the weight acts), 30 inches; weight of the weight, 83.1 pounds; weight of valve and stem, 5 pounds; weight of lever, 12 pounds; total length of lever, 32 inches? The lever is straight and parallel. Ans. 90 lb.

3. Suppose all the quantities to remain the same as in the last example, except that the valve is to blow off at 75 pounds pressure. At what distance from the fulcrum must the weight be placed? Ans. 24.58 in.

4. All quantities remaining the same as in example 3, except that the valve is to blow off at 82 pounds pressure, find the weight that must be placed on the lever. Ans. 75.1 lb.

THE STEAM GAUGE.

16. Construction.—The **steam gauge** indicates the pressure of the steam contained in the boiler.

The most common form is the **Bourdon pressure gauge,** Fig. 5. It consists of a tube *a* of elliptical cross-section, which is filled with water and connected at *b* with a pipe leading to the boiler. The two ends *c* are closed and are attached to a link, which is in turn connected with a quadrant *e*; this quadrant gears with a pinion *f* on the axis of the index pointer *g*. When the water contained in the elliptical tube is subjected to pressure, the tube tends to take a circular form, and, as a whole, straightens out, throwing out the free ends a distance proportional to the pressure. The movement of the free ends is transmitted to the pointer by the link, rack, and pinion, and the pressure is thus recorded on the graduated dial.

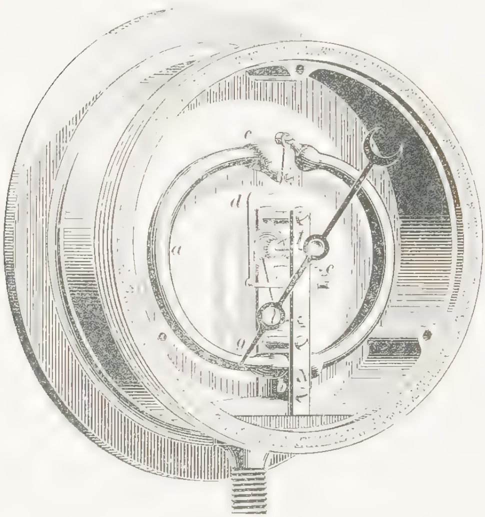


FIG. 5.

17. Methods of Graduation.—Pressure gauges for indicating steam pressure are invariably graduated to indicate pressure *above that of the atmosphere*, in pounds per square inch, and show how much the pressure has been *increased* above the atmospheric pressure. When pressure gauges are used for indicating the pressure in the condenser, they are called **vacuum gauges,** and are invariably graduated to show in inches of mercury how much the pressure has been *decreased* below that of the atmosphere. Then, to

find the absolute pressure, the vacuum-gauge reading must be subtracted from 30, and the pressure will then be given in inches of mercury. To obtain the absolute pressure in pounds per square inch, multiply the difference between the gauge reading and 30 by .49. Thus, if the vacuum gauge indicates 25 inches, the absolute pressure in the condenser is $(30 - 25) \times .49 = 2.45$ pounds per square inch.

18. Rule for Absolute Pressure.—The rule just given is entirely correct for normal atmospheric conditions at sea level. Since the pressure of the atmosphere decreases, however, with every increase of altitude, and, furthermore, since the pressure of the atmosphere at the same place is not constant, but varies between certain limits, it is better, if accuracy is desired, to use the following general rule:

Rule 4.—*To find the absolute pressure shown by a vacuum gauge, subtract the vacuum-gauge reading from the reading of the barometer and multiply the difference by .49.*

EXAMPLE.—What is the absolute pressure if the vacuum gauge indicates 19 inches, while the barometer stands at 26 inches?

SOLUTION.—Applying the rule just given, we get

$$(26 - 19) \times .49 = 3.43 \text{ lb. Ans.}$$

19. Compound steam gauges are occasionally met with in which the left-hand part of the dial indicates vacuum in inches of mercury and the right-hand part pounds per square inch *above* the atmospheric pressure. They are usually found attached to the receivers of cross-compound condensing engines.

20. Methods of Connecting Gauge to Boiler.—The gauge should be connected to the boiler in such a manner that it will neither be injured by heat nor indicate a wrong pressure. To prevent injury from heat, a so-called siphon, which may be made as shown at *a* and *b*, Fig. 6, is usually placed between the gauge and the boiler. This siphon in a short time becomes filled with condensed

steam that protects the spring of the gauge from the injury the hot steam would cause. Care should be taken not to locate the steam-gauge pipe near the main steam outlet of the boiler, since this may cause the gauge to indicate a lower pressure than really exists. In locating the steam gauge, care must also be taken not to run the connecting pipe in such a manner that the accumulation of water in it will cause an extra pressure to be shown.

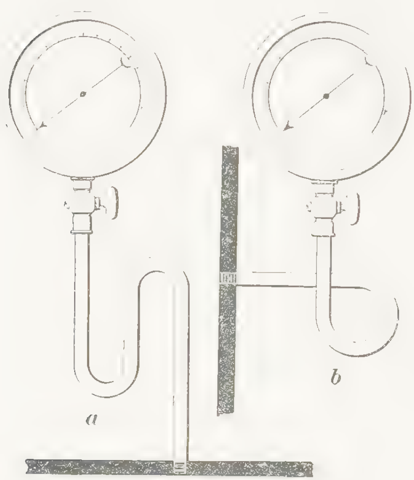


FIG. 6.

GENERAL INSTRUCTIONS FOR USE AND CARE OF STEAM GAUGES.

21. Vibration and Sticking of Pointer.—While the engine is running, it will often be noticed that the pointer of the gauge vibrates so much that the pressure cannot be read. This can be prevented by partially closing the cock below the steam gauge. The greatest of care must be used, however, to prevent an entire closing of the cock. The pointer of a steam gauge will stick occasionally and just when least expected; hence, experienced engineers always jar the gauge a little in order to dislodge any foreign matter that may be preventing movement of the pointer before they accept its indication as correct.

22. Loss of Accuracy.—Steam gauges will lose their accuracy after they have been in use for some time, owing to the spring losing its elasticity or taking a permanent set. In this case the gauge will indicate a pressure higher than the actual pressure in the boiler. This can usually be discovered by the pointer failing to return to the zero mark

when there is no pressure in the boiler. If the pressure apparently indicated when there is no pressure be subtracted from the pressure indicated when the boiler is under steam, the correct pressure will be given approximately. However, when a gauge shows a wrong pressure, a new one should be immediately substituted and the old one discarded or sent to the maker for repair.

23. Testing.—When inspecting boilers, the inspectors of boiler-insurance companies or municipal boiler inspectors, usually test all steam gauges in the plant by comparison with an accurate test gauge. The gauge to be tested and the test gauge are both attached to a vessel in which the pressure is raised by means of a small force pump, and the readings of the two gauges are compared at different pressures.

24. Checking.—As previously explained, the safety valve can be checked by means of the steam gauge when the latter is known to be accurate. Conversely, when the safety valve is known to be set correctly, the steam gauge can be checked for the blow-off pressure by watching its indication when the valve just blows off. If a steam gauge shows an error of more than five pounds, it will be condemned by most boiler inspectors. Steam gauges should be taken off at least once a month and the connecting pipe cleared by blowing steam through it. When the gauge is off, see that the hole in the nipple is perfectly clear.

25. Number of Gauges Used.—Good practice demands that one steam gauge should be attached to each boiler where there is more than one boiler used. In some regions, however, it is not uncommon to see one steam gauge do duty for a whole battery of boilers. Such an arrangement has nothing but cheapness to recommend it and is severely condemned by most engineers.

26. Repair.—With some kinds of water the spring of the steam gauge will corrode. Under no circumstances attempt to fix a corroded spring by soldering up the hole or

holes. Instead of this, send the gauge to the maker to have a new spring fitted and adjusted. When replacing the gauge after taking it off, make *sure* that the valve in the steam-gauge pipe is opened before going farther and then make sure that the gauge is operative. It has happened in numerous instances in putting up the piping with unions, that the gasket placed between the two parts of the union has been so large that in tightening the nut it has been squeezed out so as to completely stop the hole in the pipe, thus preventing the gauge from showing the pressure.

GAUGE-COCKS AND WATER GAUGES.

GAUGE-COCKS.

27. Purpose and Location.—Gauge-cocks are simple valves or cocks that are attached to the boiler for the purpose of testing the level of the water. The cocks, usually three in number, are placed either on the head or shell or they are attached to the water column. The lowest cock is placed at the lowest level that the water may safely attain and the uppermost cock at the highest desirable level. On opening a cock above the water level, steam will issue forth, and on opening one below the water level, water will appear. Hence, the level may be easily located by opening the cocks in succession. The lowest cock should be located about 3 inches above the lowest water level advisable, which, in case of a return-tubular boiler, would be 3 inches above the upper row of tubes, and in a boiler of the locomotive type about 3 inches above the highest part of the crown sheet.

GLASS GAUGES.

28. The **gauge glass** is a glass tube whose lower end communicates with the water space of the boiler and whose upper end is in communication with the steam space. Hence, the level of the water in the gauge should be the

same as in the boiler. Boilers in good work are provided with both cocks and gauges. Fig. 7 shows a common form of gauge-glass connection. The lower fitting connects with the water space and the upper fitting with the steam space of the boiler. A drip cock is placed at the lower end of the glass for the purpose of draining it. The fittings may be screwed directly into the boiler head. The gauge should be so located that the water will show in the middle of the gauge glass when at its proper level in the boiler.


FIG. 7.

WATER COLUMNS.

29. Gauge-cocks and glass water gauges connected directly to the boiler head are open to the objection that the violent ebullition at the surface of the water will cause them to indicate a wrong water level. To overcome this objection, they are frequently placed on a separate fitting known as a **water column,** which consists of a large hollow tube with its ends connecting with the steam and water spaces of the boiler far enough above and below the water level to be out of reach of the violent ebullition of the surface of the water.

30. Fig. 8 shows an arrangement of water column, gauge glass, gauge-cocks, and steam gauge that is recommended by the Hartford Boiler Insurance Company, where *H* is a round cast-iron column whose inside diameter is about 4 inches. The upper end communicates with the steam space of the boiler by means of the pipe connection *I* and the lower end with the water space through the pipe connection *J*. A drip pipe *K* is used for removing the condensed water from the column. The water glass *h* communicates with the column through the connections *L* and *M*. There are three gauge-cocks, *i*, *j*, and *k*. The center line of

the lowest one k should be located at least 3 inches above the level of the tops of the upper row of tubes in the boiler to insure that they may be always covered with water. The pressure gauge is connected to the pipe I by means of the inverted siphon pipe P, which answers the same purpose as the bends in Fig. 6.

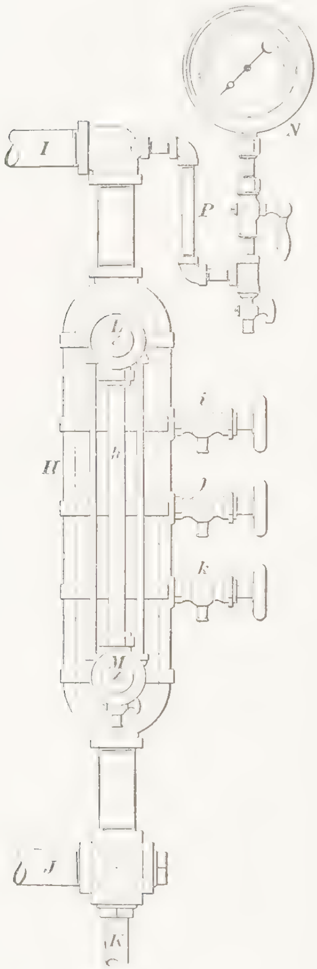


FIG. 8.

USE AND CARE OF GAUGE GLASS AND WATER COLUMN.

31. Too much reliance must not be placed on the gauge glass. If the water is muddy or contains soda, it is liable to foam, and the glass cannot give the true water level. Again, the connections between the glass and boiler may become so filled with incrustation that scarcely any water can enter the gauge. To prevent this, the glass should be blown out frequently. The water gauge and water column should be tested at least once a day. When the water gauge is attached directly to the head, open the drain cock to blow out the glass. Observe if the water returns immediately to its former level when the drain cock is closed. If if it fails to do so, this indicates that the lower fitting is choked with sediment or scale. Should the water fail to leave the glass, or leave it very slowly, it indicates that the upper fitting is choked. When this test shows the gauge to be out of order, it should be repaired at the first possible opportunity,

running in the meantime by the gauge-cocks. To remove all temptation to look at the glass, cover it with any material handy. While this may seem an unnecessary precaution, it may be the means of preventing an explosion with the consequent loss of life and property.

32. When water columns are used, they usually have a valve in each connecting pipe. To test both the gauge and the column at the same time, *double shut off* one connection and see if you get the proper fluid through the drain cock. That is, to test the water connection, shut the upper valve of the gauge glass and the valve in the steam connection. Then open the drain cock of the glass water gauge. If water issues in a constant stream, the water connection is clear. Now open the upper valve of the gauge glass and the valve in the steam connection and close the lower gauge-glass valve and the valve in the water connection. If steam flows freely from the gauge-glass drain cock, the steam passages are clear. Close the drain cock again and *open all valves.*

This method of testing is commonly expressed in a somewhat ungrammatical form as follows: *Double shut off what you get, and see if you get the other.*

The student is cautioned against the practice of testing the water column by opening the water-column drain cock only. While this will prove the water column to be clear, it will not prove the glass water gauge.

33. When the water column has no valves in the connecting pipes, test the gauge as if it were connected directly to the head, as previously explained. If the test shows the *water column* to be untrustworthy, haul the fires immediately and shut down until the column is cleared. To prevent the water column from choking up, drain it frequently and do the same with the glass water gauge. Always supplement the draining by the test given. The water gauges are the most important accessories to a steam boiler, and too much care cannot be bestowed on having them absolutely reliable.

34. The pipes for the water column should run as straight as possible and connect directly to the boiler. Under no

circumstances whatsoever should these connecting pipes be used for any other purpose or have any other pipe connection in them. When observing the glass water gauge while the boiler is working, note particularly whether the water showing in the glass is stationary or not. If the water level does not fluctuate, or pulsate up and down, as it were, it is an infallible sign that the gauge is out of order. Immediately test the gauge and water column, and if draining them fails to clear them, shut down for repairs.

35. In putting in new glasses that are held in place by rubber packing, care should be used to place the packing evenly, so that the glass will not come in contact with any part of the metal connections; on account of the unequal heat-conducting powers of the glass and metal, the glass is more likely to be broken when thus in contact with the metal than if held free by the packing.

FUSIBLE PLUGS.

36. Construction and Purpose.—In order to give warning of shortness of water in a steam boiler, fusible plugs are used. In many places they are required by law. The ordinary fusible plug in common use is shown in section in Fig. 9. It consists of a brass or iron shell threaded on the outside with a standard pipe thread. The inside is filled with some alloy having a low melting point. As long as the plug is well covered with water, the fusible metal is kept from melting by the comparative coolness of the water; but should the water sink low enough to uncover the top of the plug, the filling quickly melts and allows the steam to rush out; thus giving warning of the shortness of water. The plug shown has a conical filling, the larger end of the filling receiving the steam pressure. The conical form of the filling prevents its being blown out by the steam pressure.

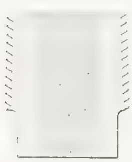


FIG. 9.

37. A good form of fusible plug is shown in Fig. 10. The plug *P* is screwed into the sheet Q, the fusible cap *R* is laid on top of it and kept in place by the nut *U*. A very thin copper cup *S* is placed over the top of the fusible cap *R* to protect it from any chemical action of the water. A valuable feature of this style of fusible plug is that when applied to the crown sheet of boilers of the locomotive type, it will give warning of shortness of water before the crown sheet is entirely uncovered, since it extends about $1\frac{1}{2}$ to 2 inches above the sheet.

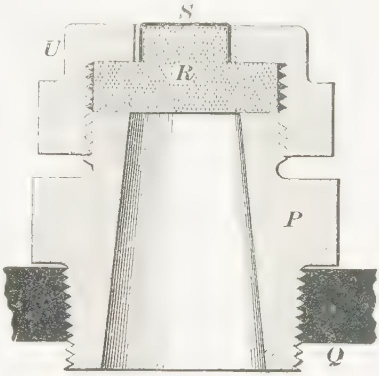


FIG. 10.

38. Location of Plugs.—In horizontal return-tubular boilers, the plug is usually placed in the back head 3 inches above the upper row of tubes. In flue boilers of the two-flue type, one plug is screwed in each flue at its highest point, or in the back head at a level about 2 or 3 inches above the top of the flues. The latter practice is considered the better, since it will give warning of shortness of water before the flues become uncovered. In firebox boilers, the plug is screwed into the highest point of the crown sheet. In vertical boilers, it is usually screwed into one of the tubes about 2 inches below the lowest gauge-cock. In water-tube boilers, it is located usually in the shell of the steam drum. In general, it should be so located that it will prevent, by the warning it gives, the overheating of the parts within the fire-line.

39. Care of Fusible Plugs.—As with other safety devices, dependence can only be placed on a fusible plug when it is given intelligent and reasonable care. It should be removed at least once a month and examined to see that the filling is not covered by hard scale. Instances are not rare when the filling has melted out and the steam been

prevented from issuing by a heavy covering of incrustation. The filling should be renewed at least every six months. Before screwing the plug home, smear a liberal supply of plumbago (graphite) on the threads; this will allow the plug to be easily removed. Do not use any oil; this will become carburized, owing to the high temperature, and will make it quite difficult to remove the plug.

40. For the filling of fusible plugs, pure Banca tin is probably the best. This can be procured almost anywhere, although, in general, it is cheaper to keep a small supply of plugs on hand, and simply replace the plug with a new one, instead of refilling it.

41. When taking charge of an old boiler, it is well to examine the fusible plug to see if it really is what it pretends to be. Instances are not rare when ignorant persons, in order not to be bothered by "leaks," have either replaced the fusible plug by a solid gas plug or stopped up the hole in the plug by driving wood or iron into it after the filling had melted out.

BLOW-OFF APPARATUS.

42. Bottom Blow-Off.—For the double purpose of emptying the boiler when necessary and of discharging the loose mud and sediment that collects from the feedwater, each boiler is provided with a pipe that enters the boiler at its lowest point. This pipe, which is provided with a valve or cock, is commonly known as the bottom blow-off. The position of the blow-off pipe has been shown in the illustrations of most the boilers described; in ordinary return-tubular boilers, it is usually led from the bottom of the rear end of the shell through the rear wall. Where boilers are supplied with a mud drum, the blow-off is attached to the drum.

43. While many boiler plants use globe valves on the blow-off pipe, their use is objectionable, since though tightly screwed down, the valve may not be properly closed on account of a chip of incrustation or similar matter getting

between the valve and its seat. As a result, the water may leak out of the boiler unperceived. Formerly, brass plug cocks were used almost entirely, which, owing to their habit of sticking tightly, were superseded by globe valves and gate valves. Within the last few years plug cocks packed with asbestos have been placed in the market, the asbestos packing removing the objectionable features of the plug cock. Many engineers now insist on the use of these cocks for the blow-off pipe. Gate valves are also used to some extent, but are open to the same objection as globe valves.

44. The bottom blow-off pipe, when exposed to the gases of combustion, should always be protected by a sleeve made of pipe, by being bricked in, or by a coil of plaited asbestos packing. If this precaution is neglected, the sediment and mud collecting in the pipe, in which there is no circulation, will rapidly become solid. Instances are not rare where the blow-off pipe has become so badly choked that on opening the blow-off cock the full steam pressure could not clear the pipe.

45. The blow-off pipe should lead to some convenient place entirely removed from the boiler house and at a lower level than the boiler. In some places the blow-off may be connected to the nearest sewer; in many localities, however, ordinances prohibiting this are in force; the blow-off is then connected to a cooling tank, whence the water may be discharged into the sewer. When the blow-off has been used for the purpose of partially emptying the boiler, the greatest care should be used to make sure that the cock or valve is properly closed. If there is a leak, it can be discovered by feeling the blow-off pipe at some distance from the boiler.

46. The Surface Blow-Off.—Boilers are often fitted with a **surface blow-off,** which is simply a pipe with a scoop-shaped fitting placed 3 or 4 inches below the water level. The pipe is provided with a cock or valve. The surface blow-off serves to remove floating impurities that would finally settle and fall to the bottom of the boiler if not removed. When using the bottom or surface blow-off to partially empty the boiler, the attendant should not leave

the boiler room under any circumstances without first closing the blow-off cock. The blow-off cock should be opened wide in order to cause a rapid flow through the pipe; this tends to prevent the lodgment of scale in the elbows and other fittings. The usual diameters of blow-off pipes are as follows: $1\frac{1}{2}$-inch pipe for boilers up to 42 inches in diameter, 2-inch pipe for diameters up to 60 inches, and $2\frac{1}{2}$-inch pipe for larger boilers.

VALVES AND COCKS.

47. For the purpose of controlling the flow of fluids through pipes, valves and cocks are universally used. Valves that allow the fluid to flow through them in either direction are divided into two general classes, viz., *globe valves* and *gate valves*.

48. Globe valves are made in a variety of forms, following the same general idea of construction. A common form is shown in Fig. 11. Here the fluid enters at A and flows out at B. The opening in the valve seat is closed by a flat removable disk, which may be

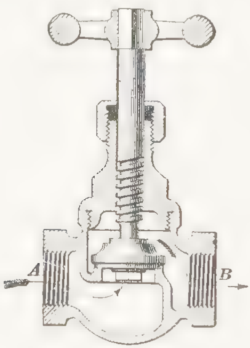


Fig. 11.

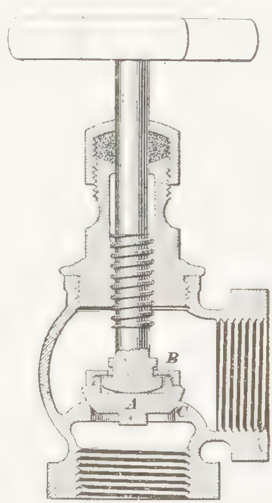


Fig. 12.

renewed when worn so as to leak. Another common construction is shown in Fig. 12. This style of valve is used at the junction of two pipes at a right angle, and hence is

termed an *angle valve.* The seat of the particular valve shown is beveled; when worn, it may be made tight again by grinding it in. Globe valves should be attached to the pipes in such a manner that the valve will close against the pressure. This will allow the valve stem to be packed without closing down the plant.

49. Gate Valves.—The waterway through a globe valve is so contorted that it obstructs the flow of a fluid through the valve to some extent. To overcome this objection, gate valves have been designed, a common form of which is shown in Fig. 13. By turning the stem *B*, the wedge-shaped disks *A* and A_1 are moved across the seats *c*, *c*, and the orifice is opened or closed gradually. The disk A_1 has cast on its lower side a projection *D* that rests on a corresponding projection *E* that is cast with the valve body. These two projections form a stop for the disk A_1; when it has come to a stop, a further turning of the stem wedges the two disks apart, pressing them tightly against the seats. A gate valve may be put on to receive the pressure on either side.

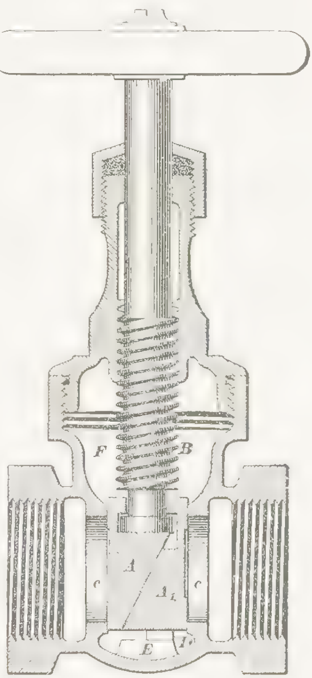


FIG. 13.

50. Check-valves are valves designed to permit the flow of fluids in one direction only and to positively prevent any return flow. The most common form of check-valve is that known as a globe check. It is shown in Fig. 14. The valve *A* is a solid disk of metal ground to the beveled seat *B*. It is guided by the wings *C* and *E* above and below the seat. The fluid passes in the direction of the arrows.

51. An improved form of check-valve, known as a **swing check,** is shown in Fig. 15. The valve disk is attached to

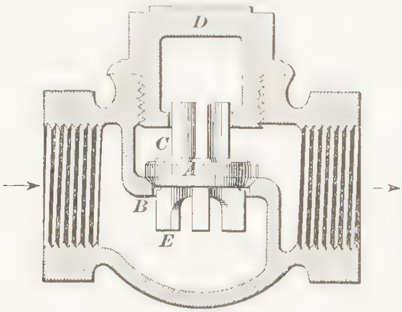


Fig. 14.

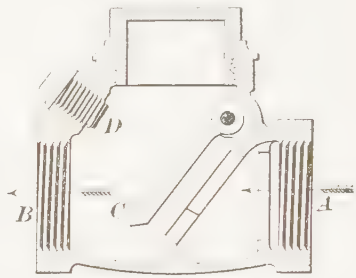


Fig. 15.

an arm that swings on a pin, as shown. The passage of the fluid through this valve is more direct than in the globe check, and the pressure required to open the valve is much less. The fluid passes through the check in the direction shown by the arrows, that is, from *A* toward *B*. In case of a rapid flow, the projection *C* on the end of the arm, to which the valve is attached, strikes against the bottom of the screw *D*, and is thus kept from going too far.

52. The two kinds of check-valves illustrated are not very well adapted for working in any other except a horizontal position. If a check-valve must be used in a vertical pipe, one made especially for this purpose should be obtained.

53. **Cocks** are rarely used for any other purpose than the blow-off pipe. Ordinary water-cocks, as used by plumbers, are not well adapted for this purpose; special blow-off cocks are made and can be obtained from all reputable makers. The objection to the ordinary plug cock is its tendency to leak around the bottom and the difficulty of moving the plug after the cock has been closed for some time. To overcome this difficulty, asbestos-packed plug cocks have been designed and are gradually coming into extensive use. These cocks have dovetail grooves cast into the body, into which asbestos is tightly driven. The

asbestos being slightly elastic, it fits snug against the plug, thus making a tight joint; at the same time, owing to the small amount of friction, it allows the plug to be turned easily. Since the asbestos is not affected by heat or moisture, it is quite durable.

DOME AND STEAM DRUM.

54. **Domes** are placed on cylindrical boilers for the purpose of increasing the steam space, and also for the purpose of drying the steam, the supposition being that the steam will be dried on account of its being farther removed from the water. The hole cut in the shell to give communication between the boiler and dome should be made only large enough to allow a man to pass through, since a large hole materially weakens the shell. The edge of the plate around the hole should be reenforced by a wrought-iron ring riveted to it. The flat top of the dome must be stayed by diagonal braces. Steam domes usually have a diameter equal to one-half the diameter of the boiler, and a height equal to about nine-sixteenths the diameter of the boiler.

55. Boilers are often fitted with a **steam drum** instead of a dome. The steam drum is simply a cylindrical vessel connected to the shell. When several boilers are set so as to form a battery, they are often connected to one drum common to all boilers. When each boiler has its own furnace, there should be a stop-valve between each boiler and the drum to allow the boiler to be taken out of service when required. When the boilers in battery have one furnace common to all of them, no stop-valve should ever be placed in the pipe connections between each boiler and the drum. Where boilers are in battery with separate furnaces, each boiler must have its own safety valve, which should always be so fitted that it cannot be cut off from the boiler under any circumstances.

56. Some boilermakers when fitting a longitudinal steam drum to a boiler will attach it by two nozzles. Many

engineers object to this method, since with an unequal expansion of the boiler and drum, which is quite likely to occur, the joints of the nozzles will become leaky, owing to the strains to which they are subjected. It is now the rule in good work to use one nozzle only. When the steam drum is used for a single boiler, its diameter may be made equal to one-half the diameter of the boiler, and its length equal to the diameter of the boiler. Where one steam drum is common to several boilers, its diameter is usually made equal to half the diameter of the boiler, and its length equal to the horizontal outside-to-outside measurement over the several boiler shells.

57. The strength of steam drums may be determined by the rules governing the strength of boiler shells. They require just as rigid inspection as the boiler itself.

THE DRY PIPE.

58. Stationary boilers are often fitted with a **dry pipe.** This consists of a pipe closed on both ends, located near the top of the shell inside of the boiler, and connected in any suitable manner to the steam pipe. The upper half of the pipe is either perforated or provided with a number of slots through which the steam enters. The supposition is that by this means most of the entrained water will be separated from the steam, the water flowing into the water space through small holes in the bottom of the dry pipe. The combined area of the perforations should be about equal to the area of the stop-valve.

THE MUD DRUM.

59. Mud drums are occasionally attached to stationary boilers for the purpose of providing a quiet place for the collection of mud and sediment in mechanical suspension in the feedwater, which is then introduced in the mud drum. It is located underneath the boiler and at the rear end, being connected to the boiler by a suitable nozzle, usually

of cast iron. Where several boilers are set in battery, they are sometimes connected to a common mud drum. This practice is permissible when the whole battery is used at once. When so fitted, none of the boilers can be temporarily taken out of service unless each nozzle is provided with a stop-valve. Owing to the difficulty of protecting the valve from the fire, this is rarely if ever done. This consideration limits the use of a common mud drum to cases where all the boilers are worked together. When a mud drum is fitted, the blow-off should be attached to it and the sediment collected in the drum frequently blown out.

MANHOLES AND HANDHOLES.

60. For the purpose of allowing the inside of the boiler to be examined, cleaned, and repaired, holes closed by suitable covers are cut into the head or shell. When of sufficient size to admit a man, they are called *manholes;* otherwise, *handholes.*

61. The Manhole.—A common construction of a manhole and its cover is shown in Fig. 16. An elliptical hole is

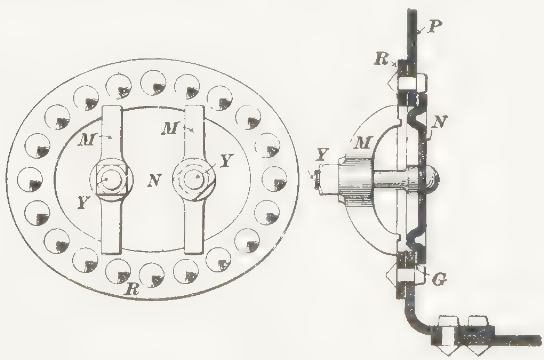


FIG. 16.

cut into the head or shell of the boiler. A wrought-iron or steel ring *R*, called a **compensation ring,** is riveted to the plate *P*, generally on the outside, for the purpose of

strengthening the plate, which is weakened considerably by the cutting of such a large hole through it. A cover N made of wrought iron, cast iron, or steel is fitted to the hole, inside of the boiler, and is provided with two studs Y, Y riveted to it. This cover is flanged and overlaps the edges of the plate about 1 inch or more all around its perimeter. A yoke M is slipped over each stud, its two extremities resting on the compensation ring. A ring G, or *gasket*, as it is commonly called, made of sheet rubber or any other pliable waterproof material, is placed between the plate and the cover and serves to make a water-tight joint.

62. Of late years it has become quite generally the practice to flange the head inwards and face its edge, thus doing away with the necessity for the compensation ring. When

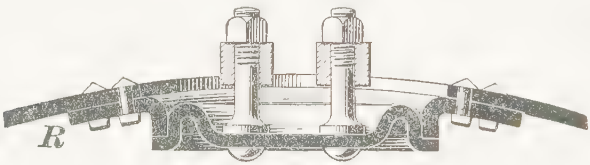


FIG. 17.

the manhole is in the shell, in the best modern practice, a flanged compensation ring is riveted to the inside of the shell, as shown in Fig. 17.

In the design shown, the edges of the ring and cover are faced and carefully fitted to each other, thus making a metallic joint. Owing to the practical difficulty of making such a joint perfectly water-tight, most engineers would prefer to place a gasket, either fibrous or metallic, between the cover and its seat, even when both are faced and fitted to each other. Manholes are usually made about 11 inches by 15 inches in the clear. If any smaller, it is a rather difficult matter for a man to get through them.

63. The Handhole.—Handholes are placed in boilers whose construction does not permit the entrance of a man,

as, for example, in vertical boilers. They are also placed in other boilers in convenient positions; thus, in boilers of the locomotive type they are usually placed in the corners of the water legs, and in horizontal return-tubular boilers are often found in the heads below the tubes. The handhole is a convenient place to rake out sediment and scale and to admit a hose for the purpose of washing out the boiler. The handhole and its cover are constructed very much like a manhole and cover; the handhole being smaller, requires but one yoke and bolt to close up the cover.

64. Manholes and handholes are made elliptical to allow the cover to be passed through the hole. The smallest diameter of the cover is somewhat less than the largest diameter of the manhole, and thus allows the cover to pass freely through the manhole. It is then turned one-quarter around inside the boiler, the gasket placed on the flange, and put in position.

65. When using sheet rubber or other fibrous gaskets, it is advisable to give them a good coating of plumbago on both sides. This will prevent their sticking to the cover and seat and allow them to be readily removed. It is rarely advisable to use the same gasket again when replacing the cover; it will usually have become carbonized by the heat and thus be too hard to make a tight joint, no matter how hard the nuts are screwed up. When the cover has been replaced with a new fibrous gasket, it is well to examine it again after steam has been gotten up and tighten up the nuts once more. A plentiful supply of graphite (plumbago) smeared on the threads of the bolts before the cover is replaced will allow the nut to be readily removed at the next examination.

66. When a manhole or handhole gasket blows out, as will happen if the work of replacing the cover has been carelessly done or the gasket cut too large, about the only thing that can be done is to haul the fire, blow out the boiler after the steam has gone down, and make the joint over again.

67. Before taking off a manhole or handhole cover, immediately after the boiler has cooled down, it is advisable to raise the safety valve or open a valve or gauge-cock so as to break the vacuum that may have been formed by the condensation of the steam remaining in the boiler. If this precaution is neglected, it may result in serious injury. While it cannot be truthfully stated that a vacuum will *always* form, instances are on record where this has happened and the cover been forced inwards by the external air pressure.

LOW-WATER ALARMS.

68. The object of the device called a **low-water alarm** is to give warning of low water in the boiler by the automatic sounding of a whistle. This object may be attained by the melting of a fusible plug or by the use of a float connected to the whistle valve. Other means, such as electrical devices and mechanical devices, using the difference in expansion of different metals for actuating the whistle valve, have been used occasionally, but have not come into extensive use, probably on account of being too delicate.

69. **Fusible-plug alarms** possess the advantage of cheapness, but they are very liable to become inoperative because of becoming incrusted with scale. If such an alarm is used, it should be cleaned once a month and the fusible plugs renewed at least every six months. When continually exposed to heat, the alloy of which fusible plugs are composed seems to deteriorate and become non-fusible, hence the recommendation to renew them occasionally.

70. Probably the great majority of alarms belong to the class using a float. They are often arranged to indicate both low and high water. A representative device of this class, known as the **Reliance high- and low-water alarm,** is shown in Fig. 18.

As shown in the figure, the device consists of two hollow floats *a*, *b* that are suspended from the bell-cranks *c*, *d*. To the short arm of each bell-crank is attached the valve stem of a small valve, there being one valve for each float. These valves serve to put the steam space of the water column in communication with the alarm whistle *e*. In this particular design of water column a sediment chamber *f* is formed at the bottom of the column that collects all foreign matter that settles from the water. The water-column drain is connected to the settling chamber.

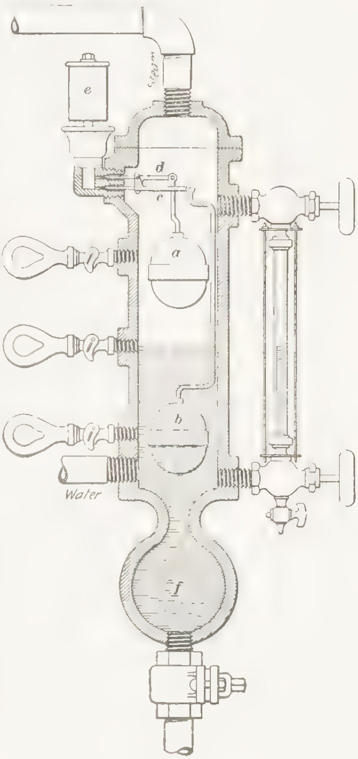


FIG. 18.

71. The operation of the alarm is as follows: when the water is at its proper level, the float *b* is surrounded by water and, being hollow, is pressed upwards. Being connected to the long arm of the bell-crank *c*, this upward pressure keeps the upper whistle valve closed. Let the water become low in the column so as to begin to uncover the float. Then, the upward pressure due to the buoyant effect of the water gradually diminishes and finally will become so small that the float will descend, thus opening the upper whistle valve and sounding the alarm. The high-water alarm float *a* keeps the lower whistle valve closed by the weight of the float. When the water rises, the float is carried upwards, the lower whistle valve is opened, and the alarm sounded.

While water alarms of all makes are perfectly reliable when in good working order, too much dependence should

not be placed on them, since, like many safety devices, they will become inoperative with long use. Float-operated alarms become inoperative through the collapsing and corrosion of the floats and the choking up of the whistle valves. Hence, these parts must occasionally be carefully looked after. The floats should frequently be examined to see if they are water-tight, since the working of the alarm depends on this fact.

THE WHISTLE.

72. Nearly every steam plant is provided with a whistle for signaling purposes. Two of the most common constructions are shown in Figs. 19 and 20. The bell, or upper

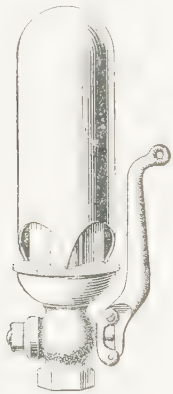


FIG. 19.

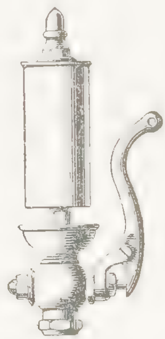


FIG. 20.

portion, is a hollow cylinder closed at the top and open at the bottom, and is held in position by a stud that passes through the center and is secured at the upper end by means of a screw and jam nut. The hollow base has a narrow circular orifice that communicates with the steam pipe and valve. As the steam rushes out of the orifice in an upward direction, toward the mouth of the bell, it slightly compresses the air contained in the bell. The air being elastic will not retain a fixed or stationary position, but will slightly spring back toward the inrushing steam, where it is again forced back in a compressed state, causing a vibration

of the air and steam. These vibrations continue as long as steam is permitted to flow and are communicated to the surrounding atmosphere, thus producing sound.

73. The tone may be changed to a higher pitch by lowering, or to a lower pitch by raising, the bell. This may be done by loosening the jam nut and turning the bell up or down, after which the nut should be again tightened.

74. Whistles are also constructed to produce two or more tones of different pitch simultaneously by dividing the bell into two or more cell-like parts, as shown in Fig. 19. Each apartment produces a different tone, and when these tones chord perfectly, the effect is quite pleasing.

75. In manufacturing establishments the whistle is usually located on the roof; that is, at a considerable distance above the boiler. In order to prevent this long pipe becoming filled with water, it is advisable to fit a small drain pipe and valve directly above the stop-valve in the whistle pipe, which is placed close to the boiler. At night the steam may then be shut off from the whistle and the drain valve opened.

When no separate valve is fitted to the whistle to shut off the steam, the blowing of the whistle, due to an accident to the whistle valve, may be stopped by pushing a stick into the bell; if this is not feasible, due to the construction of the whistle, stuff cotton waste into the bell, using a long stick.

FEEDPIPING.

76. The pipe system through which a boiler or boiler plant receives its water supply may be divided into two parts: the *external system* and the *internal system*.

The external system comprises the piping required to take the feedwater from its source of supply and to deliver it at the boiler. The internal system consists of the pipes leading from the outside of the boiler to the point of delivery.

77. The **external feed system** comprises the suction pipe of the feed-pump or injector and the delivery pipes or

feedpipes that deliver and distribute the water to the different boilers. As to the arrangement of the suction pipes, they should be as short and free from bends as it is possible to make them, especially when the water must be lifted some distance from a well or similar source of supply. In general, it is very difficult to lift water to a greater height than 24 feet at sea level, unless the pump is in excellent condition; if the water must be lifted higher, it is usually better to locate the pump farther down, excavating for it, if necessary. The suction pipes should also be perfectly airtight. When the feedwater is taken from a city water supply, it usually comes to the pump or injector under some pressure; the feed apparatus can then be located where most convenient.

78. The **arrangement of the feedpipes** naturally depends on the number of boilers to be supplied by the feed apparatus and on the extent to which the pump or injector is required to supply water to any one or all of several boilers.

79. In Fig. 21 is shown an ordinary method of arranging the feedpipes where two boilers are supplied by the same

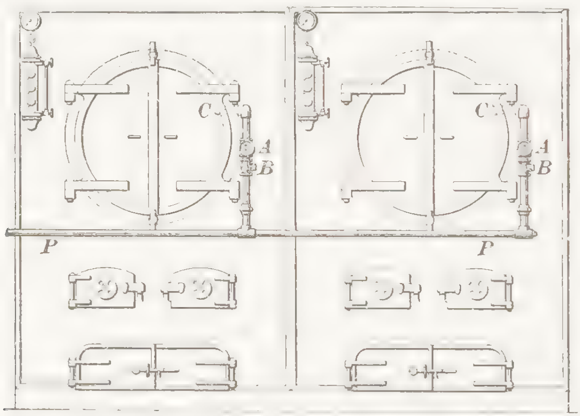


FIG. 21.

pump. The main pipe $P P$ running along the front of the boilers receives the feedwater discharged from the pump.

Each boiler is supplied by a branch from the pipe *P* entering the front head *C*. Each of these branches is provided with a globe valve *A* and a check-valve *B*. The globe valve shuts off the water from the boiler, while the check-valve allows the water to enter when the globe valve is open, but prevents its return.

80. The globe valve should always be placed nearest the boiler, thus allowing the check-valve to be examined and repaired without shutting down the boiler. With the arrangement of feedpiping shown, the feedwater can be delivered simultaneously to both boilers or to either boiler separately.

81. An arrangement of feedpiping for a plant having six boilers in two batteries with two independent feed-pumps is shown in diagrammatic form in Fig. 22. Both of the

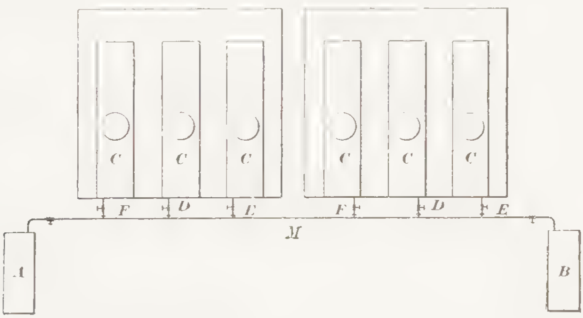


FIG. 22.

feed-pumps shown at *A* and *B* are connected to the main feedpipe *M*. This pipe has six branches, as *D*, *D*, one for each boiler. Each branch pipe has its own globe valve *E* and check-valve *F*. There are two valves in the main feedpipe, one between each pump and the nearest branch pipe. With this arrangement, either or both pumps may be used for any or all boilers. Thus, if it is desired to use pump *A* for all boilers, the valve in the main feedpipe near this pump is opened and the valve near the pump *B* is closed. The pump *A* will then deliver into any or all of the six

boilers, depending on the manipulation of the globe valves in the branch pipes.

82. When the feedwater goes through a feedwater heater before entering the boilers, it is usually advisable to provide by-pass connections, so that in case of accident to the heater, it may be cut out of service without interference with the feeding. In many plants the entire feed system is fitted in duplicate, in order to be prepared for emergencies. One system may then be supplied by injectors and the other by pumps.

83. The **internal feed system** is arranged in various ways. Its purpose is twofold: (1) to conduct the water to the proper point of discharge in the boiler; (2) to heat the relatively cold feedwater to nearly the temperature of the water in the boiler. As to the proper point of discharge, authorities differ considerably. Most engineers believe that the water should be discharged into the coolest part of the boiler and should be diffused by delivering it through a perforated pipe. Others discharge the feedwater into the steam space and use some suitable spraying device to break the entering stream into spray.

84. When discharging into the water space of horizontal tubular boilers, a common and very satisfactory arrangement is to make the feedpipe enter the front head a little below the water-line and carry it to within a few inches of the rear head. The end of the pipe is closed and a number of holes in the bottom of the pipe discharge the water downwards between the tubes. Or the water may enter through the bottom of the rear head; a horizontal pipe then carries it to within a few inches of the front head. It then passes through a vertical pipe up between the tubes to within a few inches of the water level and then returns to the rear through a horizontal pipe and is discharged downwards between the tubes. With this arrangement, the feedwater will be heated to the same temperature as the water in the boiler.

85. In cylinder boilers the feedwater usually enters the bottom of the front head. In good practice it is then carried to the rear by a horizontal pipe and discharged upwards. In some plants this is not done, however, and the water is discharged directly on the crown sheet. This is considered very poor practice by many engineers, since it will subject the plate exposed to the most intense heat to severe local stresses, which ultimately will strain the metal beyond its elastic limit and cause a rupture. In general, it is the common rule that feedwater should never be discharged on the parts of the boiler exposed to the most intense heat, nor should it be delivered in a solid stream against a plate, and, furthermore, it should be discharged in such a direction as to assist the circulation.

86. In flue boilers the water may be discharged in the same manner as in return-tubular boilers. In vertical boilers it is usually discharged into the water leg at the lowest point, although sometimes it is delivered about 2 feet above the crown sheet. In boilers of the locomotive type it may be delivered into the lower part of the cylindrical part or into the water legs below the grate. In water-tube boilers the builders always determine where to discharge the feedwater, and their advice should be followed.

FURNACE FITTINGS.

87. The **furnace fittings** consist of the *grate* with its supports, the *dead plate*, and the *bridge wall.*

88. The **grate,** which is nearly always made of cast iron, furnishes a support for the fuel to be burned and must be provided with spaces for the admission of air. The spaces and supports are alternate and are distributed evenly all over the grate surface. The combined area of all the supports is usually made nearly equal to the combined area of all the air spaces; in other words, half the grate surface is air space and half serves to support the fuel.

89. The most common type of grate is made up of single bars as *A*, Fig. 23, that are placed side by side in the furnace.

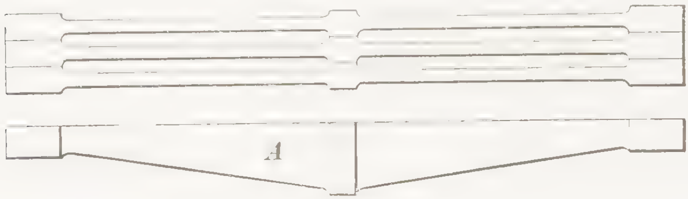


FIG. 23.

The thickness of the lugs cast on the bars determines the width of the open spaces of the grate. It is the general practice to make the thickness across the lugs twice the thickness of the support. For long furnaces the bars are generally made in two lengths of about 3 feet each, with a bearer in the middle of the grate. Long grates are generally set with a downward slope toward the bridge of about $\frac{3}{4}$ inch per foot of length. This facilitates the admission of air to the rear of the grate; it also facilitates cleaning the grate.

90. The width of the air space, and hence the thickness of the grate bar, depends largely on the character of the fuel

FIG. 24.

burned. For the larger sizes of anthracite and bituminous coals, the air space may be from $\frac{5}{8}$ to $\frac{3}{4}$ inch wide, and the grate bar may have the same width For pea and nut coal, the air space may be from $\frac{3}{8}$ to $\frac{1}{2}$ inch, and for finely divided fuel, like buckwheat coal, rice coal, birdseye coal, culm, and slack, air spaces from $\frac{3}{16}$ to $\frac{3}{8}$ inch may be used. When these small air spaces are used, the grate if made of single bars, like that shown in Fig. 23, must have the bars so thin in proportion to their length that they will warp and twist

and a large number of the bars will soon break, especially when the rate of combustion is high. To overcome this objectionable feature, the grate bar shown in Fig. 24, and known as the **herringbone** grate bar, was designed, and in many parts of the country it has almost entirely superseded the ordinary grate bar.

91. Owing to the shape of the supports for the fire, they are free to expand and contract; being quite short and of small depth in comparison to the ordinary grate bar, there is very little danger of excessive warping of the supports. In consequence, they will usually far outlast a set of ordinary grate bars. Since there are only a few large bars for the grate, it also is easier to replace a broken bar. Herringbone grate bars can be obtained in a great variety of styles and with different widths of air spaces.

92. In general, a grate bar that is suited for the kind of fuel that is to be burned should be selected. Thus, if finely divided coal is to be burned, a grate bar having small air spaces and supports should be selected, since otherwise a large percentage of the fuel will fall into the ash-pit. On the other hand, for the large sizes of coal, select bars having large air spaces, using the largest air space when caking coals are to be burned. Some varieties of bituminous coal will *cake*, that is, fuse together to a considerable degree, and the ashes and clinkers formed will be of such size that a large part of them cannot pass through the air spaces unless they are large; the grate thus becomes clogged, shutting off the air from the fire. This reduces the rate of combustion and evaporation. When putting in grate bars, they should not be fitted in tightly, but plenty of room should be given to allow them to expand.

93. Shaking Grates.—The greatest objection to stationary grate bars is that with them the furnace door must be kept open for a considerable length of time to allow the fire to be cleaned. Ashes, cinders, and clinkers will collect in the course of time on the grate, shut off the air supply, and

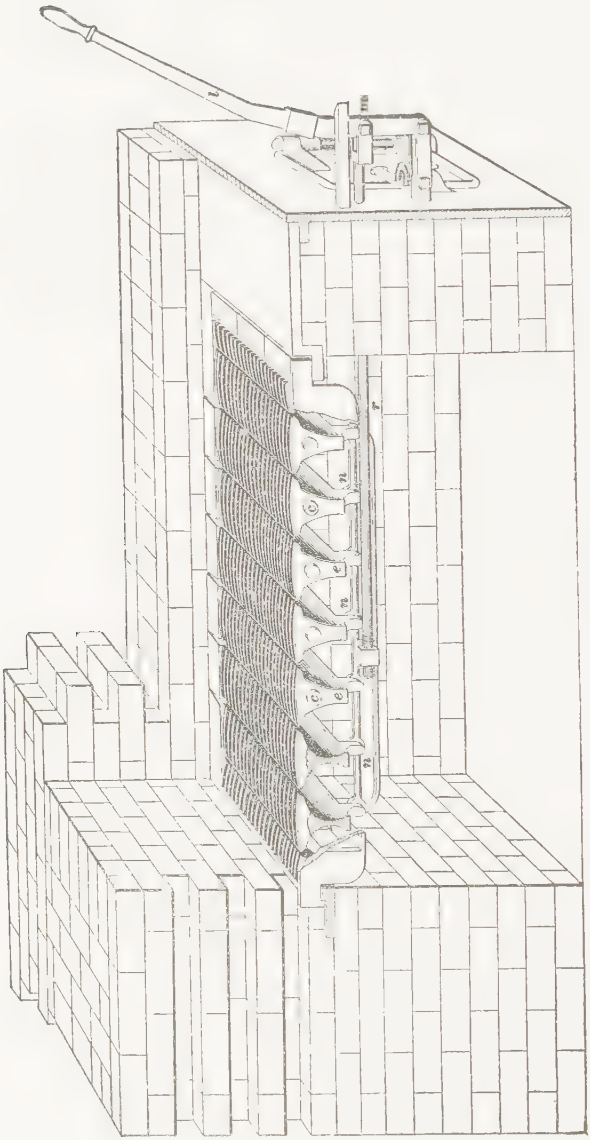


FIG. 25.

thus reduce the amount of steam generated. To restore the fire, it needs to be cleaned. Cleaning fires with a stationary grate is not only a job that severely taxes the fireman, owing to the excessive heat to which he is exposed, but also the inrush of cold air chills the boiler plates, thus producing stresses that in the course of time will crack them. To overcome these objections, grates have been designed that allow the fire to be cleaned without opening the furnace door. This is usually done by giving each grate bar a rocking motion.

94. The McClave shaking grate is shown in Fig. 25. The grate bars *c*, *c*, etc. are supported by trunnions *c*, *c* that fit into bearing bars placed on each side of the grate. A portion of the bearing bar has been removed in order to show the grate bars properly. By working the lever *l* backwards and forwards, the rod *r*, which is connected to the lever *l* by the stub lever *m*, and also with the grate bars at *n*, *n*, etc., transmits the motion of the lever to the grate bars and causes them to rotate to and fro on the trunnions.

The grate operates on the pocket principle; when the grate bars are thrown wide open they form a series of pockets that receive the ashes and clinkers; when the bars are thrown back to their upright position, the ashes are deposited in the ash-pit, and thus a quantity of ashes is removed from the bottom of the fire at each sweep of the lever.

95. This grate has been recently improved so as to allow one-half of the grate to be shaken without disturbing the other half. This is an advantage, inasmuch as the front half of the fire may be in good condition, while the back half may need shaking, and *vice versa*.

This method of cleaning is particularly applicable to anthracite coal fires, since the main body of the fuel is left undisturbed. The grate may also be used for shaking and breaking up the solid bed of fuel, as is necessary in fires of caking bituminous coals. For this purpose the lever is

vibrated rapidly back and forth through a small arc. The ashes are shaken through the grate and the mass of fuel is broken up. This grate is extensively used for burning the smaller sizes of anthracite, such as the pea and buckwheat sizes, and culm.

96. The **dead plate** is a flat iron plate usually placed just inside the furnace door. Its purpose is to furnish a support for the firebrick lining of the front of the furnace, and with stationary grates it also usually forms the support for the front end of the grate. In using some varieties of coal, it is necessary to place the coal on the dead plate and let it become heated before pushing it on to the grate.

97. The **bridge** is a low wall at the back end of the grate; it forms the rear end of the furnace. The bridge is usually built of firebrick, though in some cases it is made of wrought iron, with an interior water space communicating with the inside of the boiler. It is the office of the bridge to bring the flame in close contact with the heating surface of the boiler. The passage between the bridge and boiler shell should not be too small; its area may be approximately one-sixth the area of the grate. Likewise, the space between the grate and shell should be ample for complete combustion. Professor Thurston advises that the distance between grate and boiler shell should be one-half the diameter of the shell.

DAMPER REGULATORS.

98. In many steam plants it is required that the steam pressure be kept practically uniform. For this purpose **damper regulators** have been designed, which, operating upon a change of the steam pressure in the boiler, automatically control the position of the damper and thus regulate the volume of gases passing into the chimney. This in turn regulates the intensity of the fire and the generation of steam.

Damper regulators may be divided into three general classes:

1. Steam-actuated regulators, where the motion of a diaphragm under variation of steam pressure is transmitted directly through some multiplying device to the damper.

2. Steam-actuated regulators, where the steam in acting on a diaphragm causes a displacement of a valve, which admits steam into a cylinder, the piston of which is connected to a damper.

3. Hydraulic-operated regulators, where the movement of a diaphragm under variation of the steam pressure operates an admission valve, admitting water under pressure to a cylinder, the piston of which is connected to the damper.

99. Damper regulators of the first class are used very little, for while cheap in first cost, the regulation is rather unsatisfactory. Regulators of the second class will regulate very closely. The makers of the Curtis regulator, for instance, guarantee that the motion of the damper from one direction to the other will change with a variation of steam pressure of one-quarter of a pound per square inch, either way, from the point at which it is set to operate.

100. Regulators of the third class will also regulate very closely. Being dependent on water under pressure for their action, their application is limited to places where they can be connected either to a city water service or to a tank sufficiently high above the regulator to give the required pressure. They are sometimes connected directly to the water space of a boiler; while the damper regulator will operate when so connected, this method is open to the objection that it results in a waste of heat that may be quite large.

101. A **Spencer hydraulic damper regulator** is shown in Fig. 26. The diaphragm chamber *b* contains a flexible diaphragm dividing the chamber into two parts. The under part is filled with water which is subjected to the boiler pressure through the steam pipe *d*. The diaphragm tends to move upwards under the influence of the steam

pressure, but its upward motion is resisted by the downward force exerted by the weighted lever *c*. The weights on this lever are so adjusted that it will occupy a position midway between its two extreme positions when the steam pressure in the boiler is exactly at the point at which it is to be carried. A secondary lever *f* is hinged at *f'* to the free end of *c*; this secondary lever is fulcrumed at *m*. At *g* the valve stem of

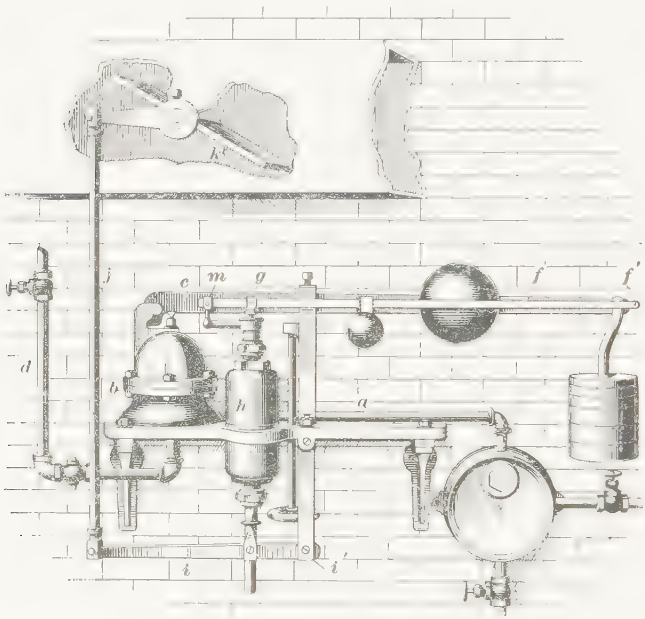


FIG. 26.

the operating valve is attached to it. This valve works inside of the piston closely fitted to the stationary cylinder *h* and serves to admit water under pressure to either side of the piston. The piston rod passes through both heads of the cylinder *h*; at its lower extremity it is connected to the lever *i* pivoted at *i'*, which, through the medium of the connecting-rod *j*, transmits any motion of the piston to the damper.

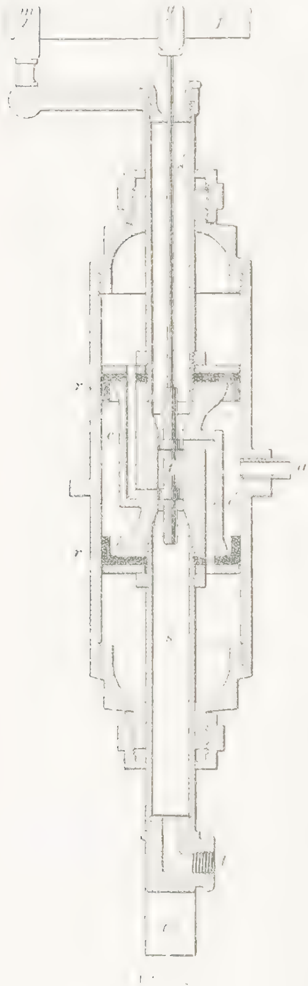
Let the steam pressure rise above that for which the damper is set. Then the diaphragm and the free end of the

lever *c* move upwards. The lever *f* being connected to *c* at one end, it swings upwards around *m* as a fulcrum; this raises the valve inside of *h* and thus admits water under pressure to the bottom of the piston in *h*. At the same time the valve places the upper side of the cylinder in communication with the water-escape pipe. In consequence thereof, the piston ascends and pulls the lever *i* upwards, which in turn rotates the damper *k*, closing it still farther. Now, as soon as the piston commences to ascend, *m* is moved upwards and the lever *f* swings around *f'* as a fulcrum. This causes the valve in the piston to move downwards in relation to the piston, thus closing the water-supply port and holding the piston in its new position.

When the steam pressure falls below the normal pressure, the levers *c* and *f* descend, and as *f* swings around *m* the valve also descends, placing the upper side of the piston in communication with the water supply and the under side in communication with the water-escape pipe. Then the piston descends and the damper opens. But *f* now swings around *f'*, and thus causes the valve to ascend in relation to the piston, which is then brought to rest.

The cylinder *h* is shown in section in Fig. 27. The piston is made water-tight by the cup-leather packing rings *r*, *r*. The water under pressure enters through the supply pipe *a* and surrounds the piston, entering through a small port into the central valve chamber and then surrounding the central port of the piston valve *t*. Let the valve move upwards. It then uncovers the ports *c'* and *c*. The water under pressure flows through *c'* into the lower part of the cylinder; at the same time the water in the upper parts flows through *c* into the hollow piston rod *s* and out at *l*. The resultant motion of the piston then returns the valve to the central position shown. If the valve descends, it admits the water into the port *c* and allows the water in the lower half of the cylinder to escape through *c'* into *s'*, which, through a by-pass port, communicates with *s*. This by-pass port, owing to the view taken, is not shown. The descent of the piston again returns the valve to its central position.

102. A damper regulator, like other automatic devices, needs intelligent supervision to have it work satisfactorily. The diaphragms will wear out in the course of time and need replacing. Care should be taken in putting it up that everything works freely and that all connections are made as directed by the manufacturer.



103. A damper regulator must not be expected to keep up a steady steam pressure without fire in the furnace; the fire should be carefully tended to. It should always be remembered that while the damper regulator will regulate the air supply to the burning fuel, it will not put fuel into the furnace.

104. Caution.—It is recommended that after shutting down the plant at night, the damper regulator be rendered inoperative. Instances are on record where failure to take this precaution resulted in running up the steam pressure to the blowing-off point during the night. Should the safety valve be out of order, all elements for a boiler explosion are present. If the regulator be left attached to the damper, the drop in the steam pressure due to banked fires will open the damper to its full extent and thus start the fires. To render the regulators inoperative, the steam may be shut off from it at night. It is considered good practice to also drain all water from it at night, especially in winter.

COMBUSTION, FIRING, AND DRAFT.

COMBUSTION.

THEORY OF COMBUSTION.

LAWS OF CHEMICAL COMBINATIONS.

1. Elements and Compounds.—Every body, every mass of matter is either an *element*, a *compound*, or a *mixture*. Iron, silver, sulphur, and oxygen are elements; water, wood, lime, and carbonic acid are compounds.

2. A compound may be decomposed or divided into separate substances. For example, if an electric current is passed through water, the water slowly disappears and two gases are formed. These gases are entirely unlike and neither resembles the water from which it is produced. Likewise, lime can be divided into two other substances—calcium and oxygen. Any substance that can thus be decomposed or divided into other substances is called a **compound.**

3. There are substances, however, that have never been decomposed into other substances. By no known process can sulphur be separated into other substances; likewise, iron, gold, arsenic, and many other substances. Substances that have never been decomposed are called **elements.**

The elements that will be considered here are the following:

Elements.	Symbols.
Hydrogen	*H*
Oxygen	*O*
Nitrogen	*N*
Carbon	*C*
Sulphur	*S*

In referring to an element it is customary to simply use the symbol, which is usually the first letter of the name. Thus, *H* stands for hydrogen, *C* for carbon, etc.

4. Chemical Combination.—When two or more elements are brought into contact under favorable circumstances, they will combine and form a new substance that is unlike either of the elements. Of course, the new substance will be a compound. Thus, if carbon and oxygen are brought together at a high temperature, they will combine and form carbon dioxide. Hydrogen and oxygen combine to form water. Hydrogen, nitrogen, and oxygen, when combined in certain proportions, form nitric acid. A given volume of nitrogen and three times that volume of hydrogen combine and form ammonia—a gas that differs greatly from both nitrogen and hydrogen.

5. It is supposed that each molecule of an element, such as hydrogen or oxygen, is composed of two atoms. It is further supposed by chemists that at a given pressure and temperature equal volumes of all *gases*, whether simple or compound, contain the same number of molecules. Thus, a cubic foot of hydrogen, a cubic foot of air, a cubic foot of steam, all contain the same number of molecules at the same temperature and pressure.

Suppose, now, that a cubic foot of hydrogen gas is allowed to come into contact with a cubic foot of *chlorine* gas (symbol, *Cl*). The mixture is exposed to heat or light and the gases combine. The process of combination is explained as follows: There is a certain attraction or affinity between the hydrogen atoms and the chlorine atoms. Under the

influence of heat or light this attraction becomes so strong that the two atoms composing the molecule of hydrogen are torn apart. Likewise, the atoms composing a molecule of chlorine separate. Each atom of chlorine seizes upon an atom of hydrogen and forms a molecule of an entirely new gas; viz., hydrochloric-acid gas. Since each atom of chlorine takes *one* atom of hydrogen, it is plain that the number of molecules of each gas must be the same. In other words, 1 cubic foot of chlorine requires 1 cubic foot of hydrogen to combine with it; these gases cannot be made to combine in any other proportion. For example, if 3 cubic feet of chlorine were placed in contact with 2 cubic feet of hydrogen, 4 cubic feet of hydrochloric-acid gas would be formed, and the extra cubic foot of chlorine would still remain chlorine. The symbol for hydrochloric-acid gas is HCl.

Suppose, now, that hydrogen and oxygen are placed in contact and heated. They will combine and form steam (or water); but it will be found that each atom of oxygen seizes 2 atoms of hydrogen to form a molecule of water, and therefore the volume of hydrogen must be double the volume of the oxygen with which it combines. This is shown by the symbol for water, which is H_2O; that is, a molecule of water is composed of 2 atoms of hydrogen to 1 of oxygen. Similarly, the symbol for ammonia is NH_3; that is, 3 atoms of hydrogen to 1 of nitrogen. Again, hydrogen and carbon form a compound; each atom of carbon seizes 4 atoms of hydrogen and forms a molecule of marsh gas. The symbol for marsh gas is, therefore, CH_4.

6. The symbol of any compound indicates how the atoms of the elements combine to form the compound. Thus, the symbol for water, H_2O, shows that 2 atoms of hydrogen and 1 of oxygen unite to form a molecule of water. The symbol H_2SO_4 (sulphuric acid) shows that 1 molecule of the sulphuric acid contains 2 atoms of hydrogen, 1 of sulphur, and 4 of oxygen.

7. Combination by Weight.—One cubic foot of hydrogen combines with just 1 cubic foot of chlorine. But upon

weighing each gas it is found that the cubic foot of chlorine weighs 35.5 times as much as the cubic foot of hydrogen. A cubic foot of oxygen weighs 16 times as much as a cubic foot of hydrogen.

It has been stated that at a given pressure and temperature equal volumes of gases contain the same number of molecules. Therefore, 1 cubic foot of oxygen must contain the same number of atoms as 1 cubic foot of hydrogen. Now, since the former weighs 16 times as much as the latter, it follows that an atom of oxygen weighs 16 times as much as an atom of hydrogen. Similarly, an atom of chlorine weighs 35.5 times as much as an atom of hydrogen. This ratio between the weight of an atom of any element and the weight of an atom of hydrogen is called the **atomic weight** of the element. The atomic weight of any element (or compound) may be found by dividing the weight of a given volume, say 1 cubic foot of the element when in a gaseous state, by the weight of 1 cubic foot of hydrogen when both are at the same temperature and pressure. The atomic weight is, therefore, much the same thing as specific gravity, except that the weight of hydrogen is used as the standard of comparison instead of the weight of water.

8. The atomic weights of the elements named above are as follows:

Hydrogen (H)	1
Oxygen (O)	16
Nitrogen (N)	14
Carbon (C)	12
Sulpher (S)	32

By the aid of these atomic weights, the composition of any substance by weight can be found when its symbol is known. Take water, symbol H_2O; that is, there are 2 atoms of H to 1 of O. Multiply the number of atoms of each by the atomic weight of the atom. Thus,

$2 \times 1 = 2$ parts by weight of hydrogen.
$1 \times 16 = 16$ parts by weight of oxygen.

18 parts by weight of water.

Then the water is composed of $\frac{2}{18} = 11.11$ per cent. of hydrogen and $\frac{16}{18} = 88.89$ per cent. of oxygen.

As another example, take carbon dioxide, CO_2. We have

1 atom of C × atomic weight, 12 = 12 parts by weight of C.
2 atoms of O × atomic weight, 16 = 32 parts by weight of O.
44 parts by weight of CO_2.

Hence, CO_2 contains $\frac{12}{44} = 27.27$ per cent. carbon, and $\frac{32}{44} = 72.73$ per cent. oxygen. From these examples, it is plain that the atomic weight of water is 18 and of carbon dioxide 44.

9. Mixtures.—Two or more substances, either elements or compounds, may be mixed together and yet not combined to form a new substance. They are then said to form a **mixture.** The mixture has the properties of the substances composing it. The most familiar example of a mixture is ordinary air. It is composed of oxygen and nitrogen, 23 parts by weight of the former to 77 parts by weight of the latter. The two gases are not combined chemically; they are simply mixed.

ELEMENTS OF COMBUSTION.

10. Combustion *is a very rapid chemical combination.* The atoms of some of the elements have a very great affinity or attraction for those of other elements, and when they combine they rush together with such rapidity and force that heat and light are produced. Oxygen, for example, has a great attraction for nearly all the other elements. An atom of oxygen is ready to combine with almost any substance with which it comes into contact. For carbon, oxygen has a particular liking, and whenever these two elements come into contact at a sufficiently high temperature, they combine with great rapidity. The combustion of coal in the furnace of a boiler is of this nature. The temperature of the furnace is raised by kindling the fire, and then the carbon of the coal begins to combine with oxygen taken from the air. The combination is so rapid and violent that a great quantity of heat is given out.

The elements that enter into combustion are oxygen, and, usually, either carbon or hydrogen. Coal, wood, and other fuels are composed almost entirely of these three elements. *Combustion is, therefore, the rapid chemical combination of oxygen with either carbon or hydrogen, or both.*

11. We have seen that when carbon and oxygen combine they form CO_2, or carbon dioxide; when hydrogen and oxygen combine they form water, H_2O. These are called the **products of combustion.** When, as is ordinarily the case, the oxygen is obtained from the air, the nitrogen of the air passes into the furnace with the oxygen. It takes no part in the combustion, but passes through the furnace and up the chimney with the CO_2 without any change in its nature; it is, however, usually called a product of combustion in air.

12. Weight of Air Required for Combustion.—It was shown that CO_2 is composed by weight of 12 parts of carbon and 32 parts of oxygen. Hence, to burn a pound of carbon requires $\frac{32}{12} = 2\frac{2}{3}$ pounds of oxygen. If the oxygen is taken from the air, it will take $2\frac{2}{3} \div .23 = 11.6$ pounds of air to supply the $2\frac{2}{3}$ pounds of oxygen. This is because only 23 per cent. of air is oxygen. The combustion of a pound of carbon may be represented as follows:

	Elements.	Products.
1 pound carbon....	1 pound carbon.....	3.67 pounds CO_2
11.6 pounds air.......	2.67 pounds oxygen ...	
	8.93 pounds nitrogen..	8.93 pounds nitrogen
12.6	12.6	12.6

That is, 1 pound of carbon requires 11.6 pounds of air for complete combustion. Of this air, 2.67 pounds is oxygen, which combines with the pound of carbon, forming 3.67 pounds of carbon dioxide. The 8.93 pounds of nitrogen contained in the air pass off with the CO_2 as a product of combustion.

Take, next, the complete combustion of 1 pound of hydrogen. The product of the combustion is water, H_2O. It has been shown that H_2O is composed by weight of 2 parts

hydrogen to 16 parts oxygen. Hence, 1 pound of H requires $\frac{16}{2} = 8$ pounds of O to unite with it. The air required to furnish 8 pounds of O is $8 \div .23 = 34.8$ pounds. The process of combustion is, therefore, as follows:

	Elements.	Products.
1 pound hydrogen..	1 pound hydrogen..	9 pounds water (H_2O)
34.8 pounds air	8 pounds oxygen ...	
	26.8 pounds nitrogen..	26.8 pounds nitrogen
35.8	35.8	35.8

13. There is one other case that may occur; the combustion of carbon may not be complete. If insufficient air or oxygen is supplied to the burning carbon, it is possible for the carbon and oxygen to form another gas, carbon monoxide, or CO, instead of carbon dioxide, CO_2.

The combustion of 1 pound of carbon to form CO, of course, requires only one-half the oxygen that would be necessary to form CO_2. This is because in CO gas 1 atom of carbon seizes 1 atom of oxygen instead of 2. To burn 1 pound of carbon to CO_2 requires 11.6 pounds of air. To burn it to CO would, therefore, require but 5.8 pounds of air.

14. The quantities of air required for combustion are shown in the following scheme:

1 Pound.	Air at 62°.	Product of Combustion.
Hydrogen.....	34.8 lb., or 457 cu. ft.	Water. Nitrogen.
Carbon burned to CO_2......	11.6 lb., or 152 cu. ft.	Carbon dioxide. Nitrogen.
Carbon burned to CO	5.8 lb., or 76 cu. ft.	Carbon monoxide. Nitrogen.

15. The fuels in common use are composed chiefly of carbon with sometimes a small percentage of hydrogen, oxygen, and incombustible matter, called *ash*. When the percentages of carbon and hydrogen are known, the air required for the combustion of 1 pound of the fuel is easily found. For example, suppose a certain coal is 90 per cent

carbon and 10 per cent. hydrogen. To burn the carbon requires $152 \times \frac{90}{100} = 136.8$ cubic feet of air; to burn the hydrogen requires $457 \times \frac{10}{100} = 45.7$ cubic feet of air. Hence, to burn 1 pound of the fuel requires $136.8 + 45.7 = 182.5$ cubic feet of air.

Rule 1.—*To find the air required to burn a given fuel: To the carbon contained in the fuel add 3 times the hydrogen, multiply the sum by 1.52, and the result will be the air required, in cubic feet, at a temperature of 62° F.*

Let A be the number of cubic feet of air required for the combustion of a given fuel; let C be the percentage of carbon and H the percentage of hydrogen, both being expressed as so many parts in 100.

Then, $$A = 152 \times \frac{C}{100} + 457 \times \frac{H}{100},$$

or $$A = 1.52\,(C + 3\,H), \text{ nearly.}$$

EXAMPLE.—The composition of a certain kind of coal is as follows:

Carbon	84 parts.
Hydrogen	5 parts.
Oxygen	7 parts.
Ash	4 parts.
	100

Find the quantity of air required to completely burn 1 pound of this fuel.

SOLUTION.—According to rule **1**,

$$A = 1.52\,(84 + 3 \times 5) = 150.48 \text{ cu. ft.} \quad \text{Ans.}$$

When the fuel already contains oxygen, a little less air is required to burn it; if it contains sulphur, a little more air will be required than given by the above rule. In either case the difference is very slight. It will be found that 1 pound of coal requires practically the same amount of air, whether it be anthracite or bituminous. Roughly speaking, it requires about 12 pounds, or 160 cubic feet, of air to burn 1 pound of carbon or coal. If less air is supplied, the combustion is imperfect; that is, the carbon burns to CO instead of CO_2.

16. Heat of Combustion.—The quantities of heat produced by the complete combustion of the elements composing the fuels have been found by experiment. They are as follows:

Hydrogen..........	62,000 B. T. U. per pound.
Carbon burned to CO_2.	14,600 B. T. U. per pound.
Carbon burned to CO .	4,400 B. T. U. per pound.

Rule 2.—*To find the heat of combustion of 1 pound of fuel, multiply the percentage of carbon by 146 and the percentage of hydrogen by 620. Add the products and the sum will be the required heat of combustion in B. T. U.*

Let B represent the B. T. U. produced by the combustion of a fuel; let C and H represent, respectively, the percentages of carbon and hydrogen composing the fuel expressed as parts in 100.

Then, $$B = 14,600 \times \frac{C}{100} + 62,000 \times \frac{H}{100},$$

or $$B = 146\,C + 620\,H.$$

EXAMPLE.—A variety of coal has the following composition:

Carbon..............................	76.5 parts.
Hydrogen..........................	4.4 parts.
Oxygen	3.0 parts.
Nitrogen...........................	1.1 parts.
Ash..................................	15.0 parts.
	100.0

How many B. T. U. will be produced by the combustion of 1 pound of this coal?

SOLUTION.—According to rule **2**,

$$B = 146 \times 76.5 + 620 \times 4.4 = 13,897 \text{ B. T. U. Ans.}$$

From the Steam Table it is found that to change 1 pound of water at 212° into steam at 212° requires 966+ B. T. U.; hence:

Rule 3.—*To find the number of pounds of water at 212° evaporated by 1 pound of fuel, divide the heat of combustion of the fuel by 966.*

EXAMPLE.—How many pounds of water at 212° can be evaporated by 1 pound of the coal of the last example?

SOLUTION.—The heat of combustion is 13,897 B. T. U.

$$13,897 \div 966 = 14.38 \text{ lb. Ans.}$$

17. Temperature of Combustion.—The theoretical temperature of the combustion of a given fuel can be easily calculated. Making no allowance for losses of heat and supposing that just enough air is furnished for the combustion, burning carbon should have a temperature about 4,940° above zero; burning hydrogen should have a temperature of about 5,800° above zero. In practice, these temperatures are never attained on account of the losses of heat. Usually, the quantity of air admitted to the furnace is from 50 to 100 per cent. more than is theoretically necessary for the combustion. This extra quantity of air enters at a temperature of 60° or 70°, and escapes up the chimney at a temperature of from 400° to 600°. A large quantity of heat is thus wasted and the temperature of the fire is greatly lowered. Where the fire is outside the boiler and the furnace is surrounded by brickwork, the furnace temperature may be 2,500° or 3,000°; but when the furnace is inside the boiler and is surrounded on all sides by water, the temperature rarely rises above 2,000° and is usually less. A high temperature is desirable, since the water of the boiler will take up heat much faster at high temperatures than at low temperatures; combustion is also more perfect at high temperatures.

EXAMPLES FOR PRACTICE.

1. How many pounds of air will be required for the perfect combustion of 7 pounds of carbon? Ans. 81.2 lb.

2. A fuel is 88 per cent. carbon and 12 per cent. hydrogen. How many cubic feet of air are required for complete combustion of 1 pound of the fuel? Ans. 188.48 cu. ft.

3. (*a*) How many B. T. U. would the combustion of the pound of fuel of example 2 give out? (*b*) How many pounds of water at 212° would 1 pound of this fuel evaporate? Ans. $\begin{cases} (a) & 20,288 \text{ B. T. U.} \\ (b) & 21 \text{ lb.} \end{cases}$

4. The chemical symbol of the product of combustion of sulphur with oxygen is SO_2 (sulphurous oxide). What is the composition of this gas by weight?

Ans. { Sulphur 50%. Oxygen 50%.

5. Assume that, with ordinary draft, double the theoretical quantity of air is used to burn a fuel. Under these circumstances, how many cubic feet of air would be required to burn 115 pounds of coal, the chemical composition being H, 5 parts; C, 90 parts; O, 3 parts; and ash, 2 parts; total, 100 parts. Ans. 36,708 cu. ft.

FUELS.

FUELS USED IN STEAM MAKING.

18. Introduction.—The fuels used in the generation of steam are chiefly coal, coke, wood, the mineral oils (such as petroleum), and natural gas. Other fuels, such as the waste gases from blast furnaces, straw, bagasse (refuse from sugar cane), dried tan bark, green slabs, sawdust, peat, etc. are also used. All these fuels are composed either of carbon alone or carbon in combination with hydrogen, oxygen, and non-combustible substances.

CLASSIFICATION OF COAL.

19. Leading Varieties.—A prominent authority, Mr. William Kent, divides coal into four leading varieties, as follows:

1. *Anthracite coal*, which contains from 92.31 to 100 per cent. of fixed carbon and from 0 to 7.69 per cent. of volatile hydrocarbons.

2. *Semi-anthracite coal*, which contains from 87.5 to 92.31 per cent. of fixed carbon and from 7.69 to 12.5 per cent. of volatile hydrocarbons.

3. *Semi-bituminous coal*, which contains from 75 to 87.5 per cent. of fixed carbon and from 12.5 to 25 per cent. of volatile hydrocarbons.

4. *Bituminous coal*, which contains from 0 to 75 per cent. of fixed carbon and from 25 to 100 per cent. of volatile hydrocarbons.

20. Anthracite coal is rather hard to ignite and requires a strong draft to burn it. It is quite hard and shiny; in color it is a grayish black. It burns with almost no smoke; this fact gives it a peculiar value in places where smoke is objectionable.

21. Anthracite coal is known to the trade by different names, according to the size into which the lumps are broken. These names, with the generally accepted dimensions of the screens over and through which the lumps of coal will pass, are given in the following table:

Culm passes through $\frac{3}{16}$-inch round mesh.

Rice passes over $\frac{3}{16}$-inch mesh and through $\frac{3}{8}$-inch square mesh.

Buckwheat passes over $\frac{3}{8}$-inch mesh and through $\frac{1}{2}$-inch square mesh.

Pea passes over $\frac{1}{2}$-inch mesh and through $\frac{3}{4}$-inch square mesh.

Chestnut passes over $\frac{3}{4}$-inch mesh and through $1\frac{3}{8}$-inch square mesh.

Stove passes over $1\frac{3}{8}$-inch mesh and through 2-inch square mesh.

Egg passes over 2-inch mesh and through $2\frac{3}{4}$-inch square mesh.

Broken passes over $2\frac{3}{4}$-inch mesh and through $3\frac{1}{2}$-inch square mesh.

Steamboat passes over $3\frac{1}{2}$-inch mesh and out of screen.

Lump passes over bars set from $3\frac{1}{2}$ to 5 inches apart.

22. Semi-anthracite coal kindles easily and burns more freely than the true anthracite coal. Hence, it is highly esteemed as a fuel. It crumbles readily and may be distinguished from anthracite coal by the fact that when just

fractured it will soil the hand, while anthracite will not do so. It burns with very little smoke. Semi-anthracite coal is broken into different sizes for the market; these sizes are the same and are known by the same trade names as the corresponding sizes of anthracite coal.

23. Semi-bituminous coal differs from semi-anthracite coal only in having a smaller percentage of fixed carbon and more volatile hydrocarbons. Its physical properties are practically the same, and since it burns without the smoke and soot emitted by bituminous coal, it is a valuable steam fuel.

24. Bituminous coal may be broadly divided into three general classes:

1. *Caking Coal.*—This name is given to coals that, when burned in the furnace, swell and fuse together, forming a spongy mass that may cover the whole surface of the grate. These coals are difficult to burn, since the fusing prevents the air passing freely through the bed of burning fuel; when caking coals are burned, the spongy mass must be frequently broken up with the slice bar, in order to admit the air needed for its combustion.

2. *Free Burning Coal.*—This is often called non-caking coal from the fact that it has no tendency to fuse together when burned in a furnace.

3. *Cannel Coal.*—This is a grade of bituminous coal that is very rich in hydrocarbons. The large percentage of volatile matter makes it valuable for gas making, but it is little used for the generation of steam, except near the places where it is mined.

25. Bituminous and semi-bituminous coals are known to the trade by the following names:

Lump coal, which includes all coal passing over screen bars $1\frac{1}{2}$ inches apart.

Nut coal, which passes over bars $\frac{3}{4}$ inch apart and through bars $1\frac{1}{2}$ inches apart.

Pea coal, which passes over bars $\frac{3}{8}$ inch apart and through bars $\frac{3}{4}$ inch apart.

Slack, which includes all coal passing through bars $\frac{3}{8}$ inch apart.

26. Lignite, according to the classification, comes under the general head of bituminous coal. Properly speaking, it occupies a position between peat and bituminous coal, being probably of a later origin than the latter. It has an uneven fracture and a dull luster. Its value as a steam fuel is limited, since it will easily break in transportation. Exposure to the weather causes it to absorb moisture rapidly; it will then crumble quite readily. It is non-caking and yields but a moderate heat, and is in this respect inferior to even the poorer grades of bituminous coal.

27. Coke is made from bituminous coal by driving off its volatile constituents. It is used chiefly for metallurgical purposes, though it is a valuable fuel for steam purposes.

28. Wood is much used in localities where it is abundant. The effective heating value of different kinds of wood differs but very little.

29. Bagasse is the refuse left after the juice has been extracted from the sugar cane by means of the mill rolls. It is used to some extent in tropical and semitropical countries. Naturally, its use is limited to the places where the sugar cane is grown.

Dried tan bark, straw, slabs, and sawdust being refuse, their use is local and usually confined to tanneries, planing and sawmills, and threshing outfits.

30. Petroleum is occasionally used as a fuel and, as such, possesses some advantages, among which are the ease of lighting and controlling the fire, the uniformity of combustion, and the economy in labor. Its disadvantages are: danger of explosion, loss of fuel by evaporation, and high price in comparison with coal. The Standard Oil Company

estimate that 173 gallons of petroleum is equal to 1 long ton (2,240 pounds) of coal, allowing for all savings incidental to its use.

31. Natural gas is abundant in parts of Ohio and Pennsylvania, and is there often used as a fuel for the generation of steam. On an average, 30,000 cubic feet of natural gas is the equivalent of 1 ton of coal.

32. Waste gases from the furnaces of rolling mills and from blast furnaces are extensively used. Naturally, their use is limited to the places where they are produced.

33. Peat may be classified as occupying an intermediate position between wood and coal. When first cut, it is totally unfit for fuel, since it contains from 75 to 80 per cent. of water. When dried, it makes a fairly good fuel.

The Babcock and Wilcox Company state that on the average 1 pound of good bituminous coal may be considered as the equivalent of 2 pounds of dry peat, $2\frac{1}{2}$ pounds of dry wood, $2\frac{1}{2}$ to 3 pounds of dry tan bark or sun-dried bagasse, 3 pounds of cotton stalks, $3\frac{3}{4}$ pounds of straw, 6 pounds of wet bagasse, and from 6 to 8 pounds of wet tan bark.

FIRING.

SYSTEMS OF HAND FIRING.

34. The style of **firing** to be adopted in any given case depends largely on the conditions present, such as the kind of fuel used, the intensity of the draft, the demand for steam, etc.

There are three methods of hand firing, known as *coking* firing, *spreading* firing, and *alternate* firing, in common use. Each of these methods has advantages peculiar to itself, and none is applicable to all cases and all conditions, that is, if economy in the generation of steam is an object.

35. The **coking system** of firing is especially adapted to bituminous coals which are rich in volatile matter. The coal is first piled on the dead plate near the door and there allowed to coke. After coking from 20 to 30 minutes the hydrocarbons will have been driven off. The coke is then pushed toward the bridge and distributed evenly over the fire. A new charge of coal is immediately heaped upon the dead plate.

This is one of the most economical methods of burning bituminous coal; if properly managed it will give excellent results in regard to the prevention of smoke. In order to get good results, the furnace door should be perforated and a suitable damper fitted for opening and closing the perforations. The air admitted in jets through the openings mixes intimately with the gases formed; the mixture passes to the rear over the bed of burning coke on the grate, where the temperature is high enough to secure their ignition and complete the combustion before they are chilled by contact with the cold surfaces of the boiler and tubes.

36. To secure success with this method, the coal should be charged in small quantities and allowed to remain on the dead plate until it is as thoroughly coked as possible; 30 minutes will, in general, be sufficient. As a matter of course, actual trial in each and every case will have to determine the proper length of time. Large lumps that will coke slowly must be broken up; if the coal cakes badly in coking, the crust thus formed must be broken with the slice bar from time to time, so as to secure the complete removal of the hydrocarbons. The size of the grate and the intensity of the draft should be such that the coke will be burned at as high a rate of combustion, per square foot of grate surface, as the conditions will permit. This results in a high furnace temperature that promotes complete combustion of the gases.

Coking firing for stationary work is but little practiced in the United States, but is much used in marine work in all parts of the world and for stationary work in European countries. It is best adapted for cases where the demand

for steam is moderately regular, since with coking firing it is somewhat difficult to force the boiler when there is a sudden heavy demand for steam. Coking firing should never be adopted for anthracite coal.

37. The **spreading system** of firing consists of covering the whole of the grate evenly with the fresh charge of coal. It is the system in most common use. While good results can be obtained by it, if the firing is done by a skilled fireman, it is not particularly to be recommended either for economical or for smokeless combustion. Best results will be obtained from the spreading system by firing light charges at frequent intervals. The habit that many unskilled firemen have of covering the incandescent coke on the grate with a thick layer of fresh coal naturally results in a lowering of the furnace temperature far below the ignition point of the hydrocarbons driven off. In consequence, there is an enormous waste of heat, and, with bituminous coal, vast quantities of black smoke are produced. To prevent this heat loss, the firing *must* be light and frequent. The spreading system is best adapted to anthracite coal in sizes larger than pea coal.

38. In the **alternate system** of firing, the coal is thrown alternately on each side of the furnace; at one firing one side of the grate is spread with coal, and at the next firing the other side receives the charge. This method is preferable to the spreading system in that the whole of the furnace is not cooled off at once by the fresh fuel. While it keeps a bright bed of fuel in at least one side of the furnace and tends to keep the average temperature of the furnace nearly constant, it cannot be recommended as being the best method for securing complete combustion of the hydrocarbons that form a valuable constituent of bituminous coal. The gases from the freshly fired coal, instead of being passed over the bright bed of fuel on the other side of the furnace, are likely to pass directly to the chimney without being sufficiently heated to secure their ignition and complete combustion.

For both bituminous and anthracite coals, it is preferable to the spreading system, however, since gas explosions in the furnace are not as likely to occur as when the spreading system is used.

39. Gas Explosions.—Explosions of the gases in the furnace, commonly called **back draft,** occur usually with small coal and are the result of careless firing. When the smaller sizes of anthracite or bituminous coal are burned with the spreading system, and when a heavy charge is put into the furnace, it is entirely feasible to obtain an explosive mixture of air and gas needing but a spark to ignite it. Owing to the interstices between the pieces of coal being small and tortuous with the smaller coals, the hydrocarbons driven off from the heavy charge are not ignited as rapidly as formed and, hence, collect and mix with the air above the grate, forming an explosive mixture if the conditions are right. All danger of a gas explosion is obviated if the firing is done very lightly, or if the alternate system is adopted, or if some part of the fire is left uncovered when putting in fresh coal, thus igniting the hydrocarbons as quickly as they are distilled off. The smaller the size of the coal, the greater is the liability of a gas explosion with a heavy charge fired spreading. With coals of sizes larger than pea coal, there is little danger of an explosion when fired spreading, except when fired thick instead of light.

DRAFT.

METHODS OF PRODUCING DRAFT.

40. The air required by the furnace for the combustion of the fuel is supplied by the draft. There are two ways of making a draft:

1. By means of a chimney.
2. By means of a fan or blower.

NATURAL DRAFT.

41. Natural draft is produced by the difference between the weight of a column of hot gases contained within a chimney and the weight of the same bulk of cold air. It is well known that any gas, when heated, is lighter, bulk for bulk, than when cool. Now, when the hot gases pass into the chimney they have a temperature of 400° or 500°, while the air outside the chimney has a temperature of from 40° to 90°. Roughly speaking, the air weighs twice as much, bulk for bulk, as the hot gases. Naturally, then, the pressure in the chimney is a little less than the pressure of the outside air. Consequently, the air will flow from the place of higher pressure to the place of lower pressure; that is, into the chimney through the furnace.

42. As an example, suppose that a chimney is 150 feet high and that the temperature of the hot gases is 500°. A column of gas at this temperature, 150 feet high, and of 1 square foot cross-section, weighs about $6\frac{1}{2}$ pounds. A column of air at 60°, of the same length and cross-section, weighs about $11\frac{1}{2}$ pounds. Hence, the difference in pressure at the bottom of the chimney is $11\frac{1}{2} - 6\frac{1}{2} = 5$ pounds per square foot. In other words, the pressure of the draft is 5 pounds per square foot.

It is customary to express the pressure of the draft in inches of water. It has been shown that the pressure of the atmosphere, 14.7 pounds per square inch, supports a column of water 34 feet high.

$$34 \text{ ft. of water} = 14.7 \text{ lb. per sq. in.};$$

or, $$34 \times 12 = 408 \text{ in. of water} = 14.7 \text{ lb. per sq. in.};$$
$$= 2{,}116.8 \text{ lb. per sq. ft.}$$

Therefore, $$1 \text{ inch of water} = \frac{14.7}{408} = .036 \text{ lb. per sq. in.};$$

$$= \frac{2{,}116.8}{408} = 5.2 \text{ lb. per sq. ft.}$$

43. The intensity of the draft is measured by means of a water gauge shown in Fig. 1. As will be seen, it is a glass tube open at both ends, bent to the shape of the letter **U**; the left leg communicates with the chimney. The air outside the chimney being heavier, it presses on the surface of the water in the right leg and forces some of it up the left leg; the difference in the two water levels *H* and *Z* in the legs represents the intensity of the draft and is expressed in inches of water.

FIG. 1.

44. The draft pressure required depends on the kind of fuel used. Wood requires but little draft, say $\frac{1}{2}$ inch or less; bituminous coal generally requires less draft than anthracite. To burn anthracite, slack, or culm, the draft pressure should be $1\frac{1}{4}$ inches of water.

The chimney is used both to create a draft and to carry off the obnoxious products of combustion. The draft produced by a chimney may vary from $\frac{1}{4}$ inch to 2 inches of water, depending on the temperature of the chimney gases and on the height of the chimney. Generally speaking, it is advantageous to use a high chimney and as low a temperature as possible for the gases.

MECHANICAL DRAFT.

45. Artificial draft is produced by means of fan blowers or steam jets. According to the manner in which it is applied, it is known as *forced draft* or *induced draft*. In a forced-draft installation, the air is forced into the ash-pit by suitable means; in an induced-draft installation, a partial vacuum is created in the chimney. Both of these systems are in common use.

46. Forced Draft.—For forcing air under pressure into the ash-pit, either a fan blower or steam jet may be used. A common construction of a fan blower is shown in Figs. 2 and 3. The shell, or housing, is made of steel plate, with a

substantial base of cast iron or wrought iron. An outlet is placed at the desired point of the circumference, whence the air is discharged into the duct leading to the ash-pit. In

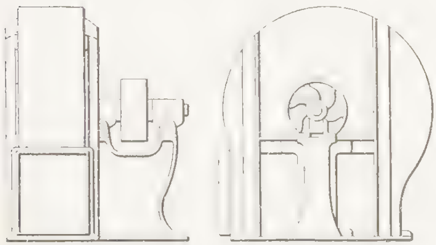


FIG. 2.

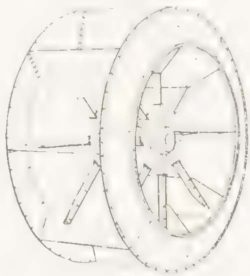


FIG. 3.

the fan shown in Fig. 2 there is one inlet, which has the fan shaft in its center and is on the side farthest from the pulley and support. The fan shaft is supported in two bearings and carries the fan wheel within the casing. The usual construction of the fan wheel is shown in Fig. 3. Arms made of T iron are fastened to the hubs and carry at their ends the blades. These blades are tied together by the side plates.

The fan is operated by a belt from an engine or may have an engine connected directly to it. The air discharged by

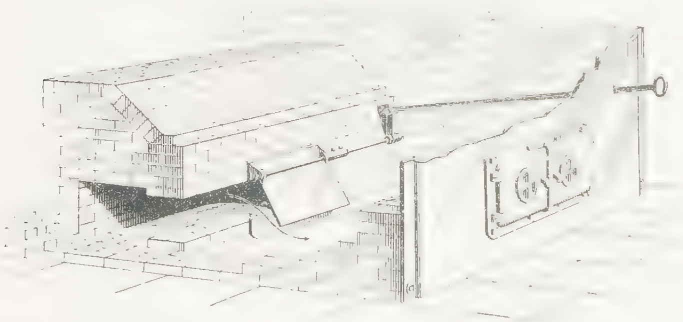


FIG. 4.

the fan may be introduced into the ash-pit through an opening in the bottom of the bridge wall, which is then built hollow, as shown in Fig. 4. The opening is closed by the

damper shown. This arrangement is recommended for a new boiler plant. When the fan draft is applied to an old plant, the air may be introduced in front through an opening in the bottom of the ash-pit, as shown in Fig. 5. When

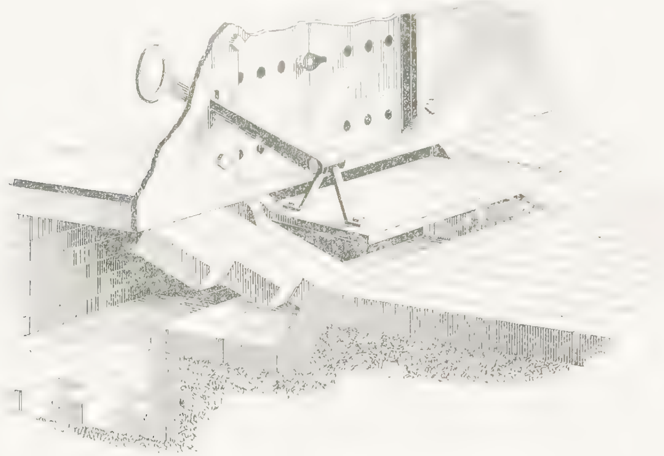


FIG. 5

the damper is closed, the ashes may be readily raked over it. The damper when opened serves to thoroughly distribute the air in the ash-pit. The ducts leading to the ash-pit may be underground, as shown in Fig. 6. There is one duct

FIG. 6

along the front of the boilers, with one branch for each ash-pit. When a low first cost is essential, galvanized-iron pipes may be used for ducts, with the main pipe overhead and a branch pipe extending down to each boiler. While the

installation of a fan blower is more costly than that of a steam jet, it is much more economical in running than the latter.

47. A **McClave argand steam blower** for forcing air into the ash-pit is shown in Fig. 7. The blower consists of

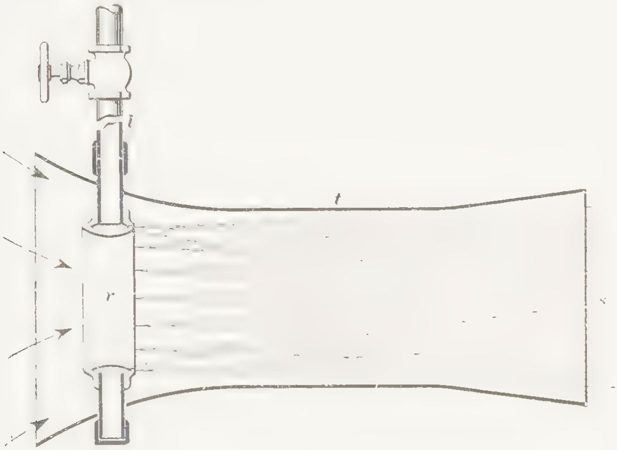


FIG. 7.

a long air tube *t* discharging from the end *s* below the grate. In the other end of the tube is placed a ring-shaped tube *r*, perforated on the right with small holes. Steam from the boiler is led into the ring by the pipe *l* and escapes in jets through the perforations, carrying air along with it into the ash-pit. This method of producing a forced draft is extensively used in the coal regions for burning the finer grades of anthracite coal and for bituminous slack.

48. Induced draft, or **suction draft,** as it is occasionally called, may be produced by a fan placed in the uptake. A typical induced-draft installation, in which the engine is connected direct to the fan, is shown in Fig. 8.

The fan, when running, creates a partial vacuum in the uptake and furnace, thereby causing the outside air, under the influence of its greater pressure, to flow into the ash-pit. The fan may be placed directly over the boilers, as shown in

Fig. 8, and discharge the gases into a short steel stack; or, it may be placed where most convenient. When applying

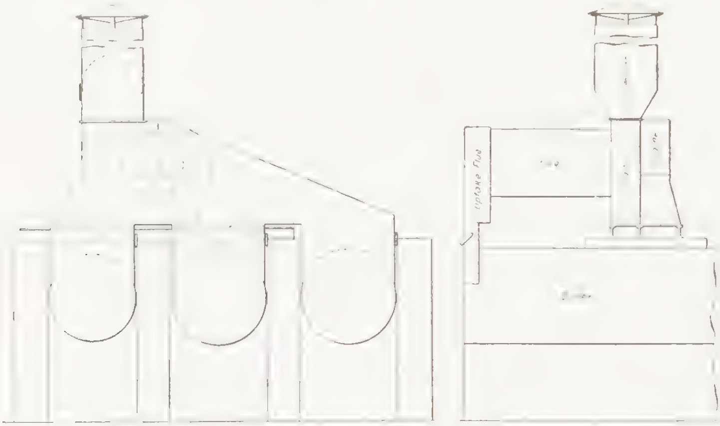


FIG. 8.

induced-fan draft to an old plant using natural draft, the fan may be placed in connection with the smoke flue in such a manner as to draw the gases from it and return them to it again at a point farther on, whence they pass to the existing chimney. Suitable dampers serve to switch off the fan and pass the gases directly through the flue when the fan is not in service.

A vacuum may also be caused in the chimney by means of a steam jet; this is a method commonly adopted in locomotives, in which the exhaust nozzle is placed in the stack, and the exhaust steam, by carrying with it the air in the stack, creates the draft. This method is seldom used in stationary practice.

ADVANTAGES OF MECHANICAL DRAFT.

49. Mechanical draft possesses certain advantages over natural draft which in many cases make it advisable to substitute it for natural draft. These advantages are as follows:

50. Adaptability.—Blowers may be applied under almost all circumstances and are independent of location.

They may be used to assist an existing chimney in order to help a plant weak in chimney power or primarily for the creation of draft.

51. Ease of Control and Flexibility.—A fan may be automatically controlled to produce a very close approach to a uniform steam pressure by attaching an automatic damper regulator to the throttle valve of the engine. With natural draft, the intensity of the draft depends on the intensity of the fire, and is, therefore, least when the fire is low. With mechanical draft, however, the draft is independent of the condition of the fire, and, consequently, banked fires may be started up quickly. Furthermore, it can be readily adjusted for the combustion of different kinds of fuel and for widely varying combustion rates. Owing to the intensity of draft that may be created, not only can the low grades of coal be burned successfully, but, also, the amount of steam generated by each boiler may be greatly increased.

52. Independence of Climatic Conditions.—As is well known, there is a decided difference in the intensity of the draft produced by a chimney in summer and that produced in winter time. Likewise, on damp and muggy days, the draft is lessened. But mechanical draft is entirely independent of these conditions.

53. Forced draft produced by a steam blower possesses the advantage of cheapness in first cost when applied to a few boilers. For a large installation its cost will probably exceed that of a fan-draft installation, since one steam blower will be required for each ash-pit. Furthermore, the steam consumption of the steam blower is quite high. It seems to be an advantage when used with coals that clinker badly, some engineers reporting that it greatly lessens the trouble from clinkers adhering to the firebrick lining of the furnace.

The draft produced by a steam blower is as flexible and as easily adapted as fan draft. It is, however, difficult to regulate it automatically so as to get satisfactory results.

BOILER DESIGN.

BOILER MATERIALS.

1. The materials used in boiler construction are *wrought iron*, *steel*, *cast iron*, and to some extent *copper* and its alloys. The qualities required of boiler material are: (1) Ductility, in order that it may undergo, without injury to its tensile strength, the various processes to which it must be subjected in being made up into boiler parts. (2) Tensile strength, to resist the stresses due to the steam pressure. (3) Toughness and elasticity.

WROUGHT IRON.

2. Wrought iron was, until a few years ago, almost the only material of which boiler shells, fireboxes, and tubes were made. The wrought iron used in the better grade of work is known commercially as *C H No. 1 flange iron.* This has a tensile strength of from 50,000 to 65,000 pounds per square inch and is quite homogeneous; that is, it has an even texture. This grade of iron is not very fibrous, does not blister nor crack much in the fire, stands repeated heating, and flanges well. When the iron is made to have the higher tensile strength, it is usually found to be obtained at the sacrifice of ductility.

3. As a measure of ductility, good boiler iron should show an elongation of at least 20 per cent. in a length of 8 inches, that is, when a bar 8 inches long is placed in a

testing machine, it should stretch $8 \times \frac{20}{100} = 1\frac{6}{10}$ inches before breaking. Iron used for rivets should be of the very best quality; it should be soft and tough; the tensile strength should not exceed 50,000 pounds per square inch; and a good rivet should bend double while cold without fracture.

STEEL.

4. Steel is now rapidly supplanting iron as a boiler material. The steel used in boiler construction belongs to the class known as soft or mild steel and is nearly always made by the Siemens-Martin process, more commonly called the "open-hearth process." It exceeds iron in tensile strength, thus permitting the use of thinner sheets; it is as ductile as and more homogeneous than iron, and by the modern processes it can be manufactured more cheaply than iron.

TESTS.

5. Methods of Testing.—The suitability of a given piece of material for boilermaking is determined by means of a number of tests, some of which are chemical and some physical.

6. The **chemical test** or *analysis* shows whether the specimen contains the right proportions of the elements required to secure the strength and other useful properties called for by the conditions under which the material is to be used; it also shows whether the material is sufficiently free from the elements whose effect on the material has been shown by experience to be bad. For example, it has been found that a certain percentage of carbon in steel will secure a given degree of strength and hardness, while more than very small percentages of phosphorus and sulphur will render the metal useless—sulphur, because it makes the steel *hot short*, that is, brittle and difficult to work

when hot, and phosphorus, because it makes the steel *cold short*, or brittle, when cold.

7. The **physical tests** to which steel and iron for boiler-making are subjected may be divided into two general classes, the **tensile** test and the **bending** test.

8. For the **tensile test,** strips are cut from the plates as they come from the rolls, or pieces of the rolled rivet rods form the specimens on which tests are made. In order to secure uniform and reliable results, it is important to have the specimens prepared in a careful and uniform manner.

Fig. 1 shows the form and general dimensions of the specimens for tests of plate that are prescribed by the

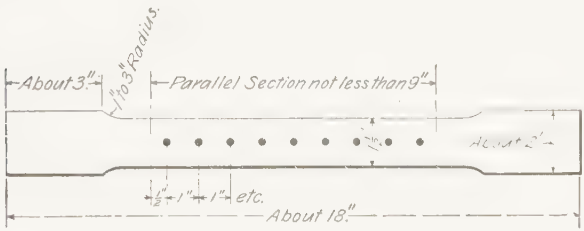


Fig. 1.

"Manufacturers' Standard Specifications." The strip from which the specimen is prepared is cut from a part of the plate that will show a good sample of its general quality and the specimen is carefully given the form shown, either by planing or milling the edges. The specimen is divided along its length for a distance of 8 inches into spaces 1 inch long, the division marks being made by a very light center-punch mark.

The test consists of gripping the ends of the specimen in a machine called a *testing machine* and pulling it until it breaks. The machine weighs the pull, and by this means the amount of pull or the *load* required to break the specimen is determined. During the process of pulling, observations are also made for determining the elastic limit, that is, the load at which the specimen begins to have a

permanent set. Since the exact determination of the elastic limit requires very careful and delicate measurements of the length of the specimen as it stretches with an increase of load, the **yield point,** that is, the load under which the specimen begins to change in length rapidly without a corresponding increase in the load, is often accepted as the elastic limit.

During the process of pulling, the specimen stretches considerably before it breaks. The amount of this stretch is determined by placing the broken ends together and measuring the distance between two of the punch marks. If the two marks were 8 inches apart before the piece was pulled and after breaking they are found to be 10 inches apart, the specimen has stretched 2 inches, or 25 per cent., in the original 8 inches. The amount of stretch, or the *elongation*, as determined in this way, is a valuable indication of the ductility of the material and its ability to withstand sudden shocks and such processes as flanging without serious injury.

At the point of fracture the specimen draws down, so that its sectional area is considerably less than the sectional area before being pulled. The reduced area is sometimes carefully measured and compared with the area before pulling; this determines what is known as the *reduction in area*, which is also a useful indication of the ductility of the material.

9. The **bending test** for steel boiler plate or rivets consists in bending a specimen double and closing it down tight. Two bending tests are often prescribed. In one, the specimen is bent cold without any treatment after it comes from the rolls. In the other, the specimen is heated to a bright red and then quenched, or suddenly cooled, in water having a temperature of about 80°. The former is called a **cold bend** and the latter a **quench bend.** A good quality of steel for boiler plates or rivets should bend flat on itself, either cold or quenched, without any indication of cracking on the outside of the bend.

10. Coupons.—In some cases, the plates to be used for boilermaking are not sheared to the exact size required at the mills, but are furnished to the boilermaker with a strip which can be cut from the plate and tested at the boiler shop. In this way, the user of the plate can satisfy himself that the quality of the plate conforms to the specifications. The strip that is to be cut from the plate for testing is called a **coupon.**

11. Stamping. — Many specifications require boiler plates to be stamped with the name of the maker of the plate and its tensile strength in pounds per square inch. This stamp should be on a part of the sheet that will enable it to be readily seen after the boiler is made. The United States laws governing the construction and inspection of marine boilers specify that every iron or steel plate intended for the construction of boilers to be used on steam vessels shall be stamped by the manufacturer in the following manner: "At the diagonal corners, at a distance of about 4 inches from the edges and at or near the center of the plate, with the name of the manufacturer, the place where manufactured, and the number of pounds tensile strain it will bear to the sectional square inch. *Provided, however,* that where butt straps are used, the stamps in corners shall be extended to a distance not to exceed 8 inches from the edges."

SPECIFICATIONS FOR BOILER STEEL.

12. The "Manufacturers' Standard Specifications" divide boiler steel made by the open-hearth process into four grades, assigning to each grade certain physical and chemical properties.

1. *Extra-Soft Steel.*—This steel has an ultimate tensile strength of 45,000 to 55,000 pounds per square inch; an elastic limit of not less than one-half the tensile strength; and an elongation of 28 per cent. in 8 inches. Either cold or quenched, it must stand a bending of 180° flat on itself

without showing any fracture on the outside of the bent portion. Not more than .04 per cent. of phosphorus and .04 per cent. of sulphur must be contained in the steel.

2. *Firebox Steel.*—The ultimate tensile strength ranges between 52,000 and 62,000 pounds per square inch; it must have an elastic limit of not less than one-half the ultimate tensile strength and an elongation of 26 per cent. in 8 inches. It must stand the same bending test as the extra-soft steel. As implied by the name, this grade of steel is especially made for fireboxes. The chemical properties prescribed are that the percentage of phosphorus must not exceed .04 per cent., nor must there be more than .04 per cent. of sulphur.

3. *Flange Steel or Boiler Steel.*—The ultimate tensile strength and the elastic limit are the same as that of firebox steel, but the elongation required is slightly less, being 25 per cent. in 8 inches. It must not contain more than .06 per cent. of phosphorus and .04 per cent. of sulphur. It must also stand the same bending test as the other grades. Flange steel is used for boiler shells and boiler heads.

4. *Rivet Steel.*—This grade of steel must conform to all the requirements given for extra-soft steel.

13. The specifications adopted by the American Boiler Manufacturers' Association in 1898 provide for the following physical and chemical properties of steel:

1. Shell plates *not* exposed to the direct heat of the fire or gases of combustion, as the external shells of internally fired boilers, may have from 65,000 to 70,000 pounds tensile strength; elongation not less than 24 per cent. in 8 inches; phosphorus not over .035 per cent.; sulphur not over .035 per cent.

2. Shell plates in any way exposed to the direct heat of the fire or the gases of combustion, as the external shells or heads of externally fired boilers, or plates on which any flanging is to be done, to have from 60,000 to 65,000 pounds

tensile strength; elongation not less than 27 per cent. in 8 inches; phosphorus not over .03 per cent.; sulphur not over .025 per cent.

3. Firebox plates, or such plates as are exposed to the direct heat of the fire, or flanged on the greater portion of their periphery, to have 55,000 to 62,000 pounds tensile strength; elongation 30 per cent. in 8 inches; phosphorus not over .03 per cent.; sulphur not over .025 per cent.

For all plates, the elastic limit to be at least one-half of the ultimate tensile strength.

4. *Bending Test.*—Steel up to ½ inch in thickness must stand bending double and being hammered down on itself; above that thickness it must bend around a mandrel having a diameter equal to one and one-half times the thickness of plate and down 180° without showing signs of distress.

5. Rivets must be of good charcoal iron, or of a soft, mild steel, having the same physical and chemical properties as firebox plates.

6. Boiler tubes to be of charcoal iron or mild steel, lap-welded or drawn; to be round, straight, free from scales, blisters, and mechanical defects; each tube to be tested to 500 pounds internal hydrostatic pressure. All tubes must stand hammering down flat and expanding, as well as flanging over, without flaws, cracks, or opening of the weld.

7. Braces, stays, and staybolts to be made of iron or mild steel, iron to have a tensile strength of not less than 46,000 pounds and an elastic limit of not less than 26,000 pounds; elongation not less than 22 per cent. for bolts of less than 1 square inch area, nor less than 20 per cent. for bolts of 1 square inch area or more, in a length of 8 inches. Steel to have a tensile strength not less than 55,000 pounds; an elastic limit of not less than 33,000 pounds; an elongation of not less than 25 per cent. for bolts of less than 1 square inch in area, nor less than 22 per cent. for bolts 1 square inch or more in area.

8. The material for staybolts must stand the following bending test: a bar threaded with a sharp die having a

V thread with rounded edges must bend cold 180° around a bar of same diameter without showing any cracks or flaws. The material for stays and braces must be fully equal to staybolt stock.

SPECIAL BOILER MATERIALS.

14. Cast iron is used only in sectional boilers like the Harrison, and in these cases steel castings are often substituted for it. Cast iron has some advantages as a boiler material. It is cheap, durable, and will withstand corrosion better than wrought iron or steel. Its brittle and treacherous nature, however, prevents its use in high-pressure boilers, except for mountings and settings. Sometimes, however, the ends or heads of plain cylindrical and flue boilers are made of heavy plates of cast iron.

15. Copper is used in England for locomotive fireboxes and staybolts. Iron and steel are used for that purpose in the United States.

16. Brass was formerly used in the construction of tubes. At present, it is used only in the construction of some of the fittings.

DESIGN OF RIVETED JOINTS.

EFFICIENCY OF JOINTS.

17. Explanation of Efficiency.—When designing a riveted joint, it should be the aim of the designer to distribute and proportion the rivets in such a manner that the strength of the joint will be as nearly equal to the strength of the solid plate as is practicable.

The ratio between the strength of the plate and the strength of the joint is called the **efficiency of the joint,** and is commonly expressed as a percentage of the strength of the solid plate.

18. When calculating the efficiency of a joint, it is customary to consider the plate as being divided into a number of equal strips at a right angle to the joint. Conceiving the riveting to be divided into equal groups, the width of the equal strips is to be such as to enclose one group of rivets.

In Fig. 2, the lines *A B* and *C D* indicate the width of the strip containing a group of rivets. For single-riveted,

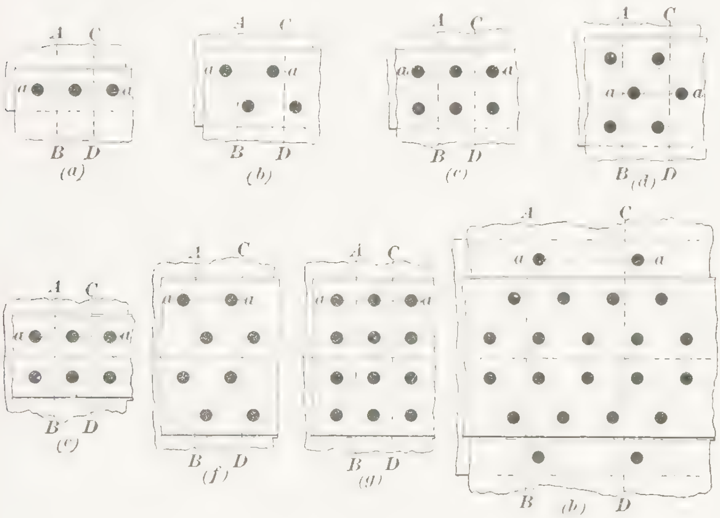


FIG. 2.

double zigzag-riveted, and double chain-riveted lap joints and butt joints, and triple-riveted lap joints, the width of the strip will be equal to the pitch of the rivets. In a triple-riveted butt joint, with two cover-plates and every alternate rivet left out in the outer row, as shown in Fig. 2 (*h*), the width of the strip is to be taken as being equal to the pitch of the rivets in the outside row.

In order to determine the efficiency of the joint, its resistance must be computed for each of the different ways in which it may fail. Then, if 100 times the lowest resistance of the several parts of the joint be divided by the resistance of the solid plate, the efficiency of the joint expressed in per cent. will be obtained.

19. Failures of Riveted Joints.—A riveted joint may fail in one of several ways. (1) The plate may break along the rivet holes, as shown at *a* in Fig. 3. This shows the

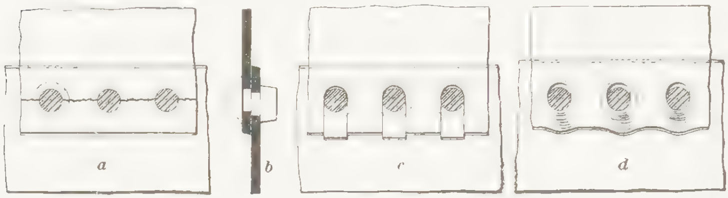


FIG. 3.

rivets to be stronger than the plate. (2) The rivets may shear off, as shown at *b*. This indicates that the plate is stronger than the rivet. (3) The plate in front of the rivet may shear out, as shown at *c*. (4) The plate may crush in front of the rivet. Experiments show that when the different kinds of joints fail by breakage of the plates, this breakage occurs most often in the manner shown in Fig. 2; that is, along the line *a a* through the rivet holes. It is evident that the rivet hole has the effect of reducing the area of the section of the strip by an amount equal to its diameter multiplied by the thickness of the plate. The difference between the original cross-sectional area of the strip and the product obtained by multiplying the diameter of the rivet hole by the thickness of the plate is the *net area* or the *net section* of the strip.

20. Computing the Resistance.—Double zigzag-riveted joints have been known to fail along lines drawn from the center of one rivet in one row to the center of the next rivet in the adjacent row. With the rivet spacing ordinarily used, the liability of the joint to fail in this manner is small; and as it is a rather difficult matter to calculate the resistance to failure along these diagonal lines, it is customary to disregard it and calculate the resistance of zigzag-riveted joints on the assumption that failure will take place along the line *a a*, Fig. 2.

Let R = resistance of solid strip to a tensile stress;
R_n = resistance of net section of strip to a tensile stress;
R_s = total resistance of rivets in strip to shearing stress;
t = thickness of strip in inches;
w = width of strip;
N = number of rivets resisting shear;
d = diameter of rivet hole;
p = pitch of rivets;
T = ultimate tensile strength per square inch of section;
S = ultimate shearing strength per square inch of section.

The resistance of the solid strip to a tensile stress is given by the following rule:

Rule 1.—*Multiply the ultimate tensile strength per square inch of the material by the thickness of the plate and the width of the strip.*

Or, $$R = T t w.$$

The resistance of the net section of the plate to a tensile stress is given by the following rule:

Rule 2.—*From the width of the strip subtract the diameter of the rivet hole. Multiply the remainder by the thickness of the plate and its ultimate tensile strength in pounds per square inch.*

Or, $$R_n = (w - d) t T.$$

21. When the joint is subjected to a tensile stress, there will be a tendency to shear the rivets. In lap joints and butt joints with single cover-plates, the rivets are in single shear; in butt joints with double cover-plates, in double shear. In triple-riveted butt joints designed as shown in Fig. 2 (*h*), where the inside cover-plate is wider than the outside cover-plate, the two inside rows of rivets are in double shear and the outside row in single shear.

The student is cautioned against considering a butt joint like that shown in Fig. 2 (*e*) as a double-riveted joint. The butt joints are, respectively, single-riveted in Fig. 2 (*e*); double zigzag-riveted in Fig. 2 (*f*); double chain-riveted in Fig. 2 (*g*); and triple-riveted in Fig. 2 (*h*). That the joint shown in Fig. 2 (*e*) is single-riveted follows from the fact that the separation of one plate from the other is opposed by only one row of rivets.

22. The shearing resistance of rivets in double shear is not twice their resistance in single shear, as often assumed. Experiments on steel and iron rivets at the Watertown arsenal have shown that their resistance in double shear averages 1.85 times their resistance in single shear.

The shearing resistance of the rivets in the strip considered may be calculated as follows:

Rule 3.—*Multiply the area of the rivet hole by the number of rivets resisting the shear in the strip considered. Multiply this product by the ultimate shearing strength of the rivet material. When the rivets are in double shear, multiply again by 1.85.*

Or, $R_s = d^2 \times .7854\ N\ S$ for single shear,

and $R_s = d^2 \times .7854\ N\ S \times 1.85$ for double shear.

The attention of the student is called to the fact that in calculating the shearing resistance of the rivets, the *area of the rivet hole* is used. It is the usual practice to make the rivet hole $\frac{1}{16}$ inch larger than the rivet. When the rivet is driven, it fills this hole. Then, its new area, which equals the area of the hole, in conjunction with the shearing strength of the material, determines the shearing resistance.

23. Considering the case of failure by shearing out and crushing in front of the rivet, experiments have shown that with the rivet spacing and proportions generally used, there is no danger of the joint failing in this manner if the distance from the center of the rivet to the edge of the plate is made $1\frac{1}{2}$ times the diameter of the rivet hole.

By applying rules **1**, **2**, and **3**, the efficiency of a joint may be investigated. First of all determine the resistance of the solid plate; then determine the resistance of the net section to tensile stress; and, finally, the shearing resistance of the rivets. Then find the efficiency as shown in Art. **18.**

EXAMPLE 1.—In a single-riveted lap joint, as shown in Fig. 2 (*a*), the pitch of the rivets is 2 inches. The rivet holes are $\frac{7}{8}$ inch in diameter. The plate is $\frac{3}{8}$ inch thick and has an ultimate tensile strength of 55,000 pounds per square inch of section. Taking the ultimate shearing strength of the rivet material at 38,000 pounds per square inch of section, what is the efficiency of the joint?

SOLUTION.—According to the statement in Art. **18**, a strip 2 inches wide is to be considered. By rule **1**, its resistance to a tensile stress is $55,000 \times 2 \times \frac{3}{8} = 41,250$ pounds. By rule **2**, the resistance of the net section to a tensile stress is $(2 - \frac{7}{8}) \times 55,000 \times \frac{3}{8} = 23,203$ pounds. As there is one rivet in single shear in the strip, its shearing resistance, by rule **3**, is $(\frac{7}{8})^2 \times .7854 \times 1 \times 38,000 = 22,850$ pounds. Comparing the two resistances, it is seen that the joint will fail by shearing the rivet. The efficiency of the joint is

$$\tfrac{22850}{41250} \times 100 = 55.4 \text{ per cent., nearly. Ans.}$$

EXAMPLE 2.—In a triple-riveted lap joint, as shown in Fig. 2 (*d*), the pitch of the rivets is $3\frac{1}{4}$ inches. The rivet holes are $\frac{13}{16}$ inch in diameter. The plate is $\frac{3}{8}$ inch thick and has an ultimate tensile strength of 60,000 pounds per square inch of section. The ultimate shearing strength of the rivet iron is 38,000 pounds per square inch of section. Find the efficiency of the joint.

SOLUTION.—According to the statement in Art. **18**, a strip $3\frac{1}{4}$ inches wide is to be considered. The resistance of the solid strip to a tensile stress, by rule **1**, is $60,000 \times 3\frac{1}{4} \times \frac{3}{8} = 73,125$ pounds. By rule **2**, the resistance of the net section to a tensile stress is $(3\frac{1}{4} - \frac{13}{16}) \times 60,000 \times \frac{3}{8} = 54,843.7$ pounds. There being three rivets in single shear in the strip to resist the shearing stress, the shearing resistance, by rule **3**, is $(\frac{13}{16})^2 \times .7854 \times 3 \times 38,000 = 59,107.5$ pounds. The calculations show that the net section of the plate is weakest. The efficiency of the joint is

$$\frac{54,843.7}{73,125} \times 100 = 75 \text{ per cent. Ans.}$$

EXAMPLE 3.—Find the efficiency of a triple-riveted butt joint with two cover-plates, having every alternate rivet left out in the outer row. The outer row of rivets is in single shear and the inner rows are in double shear. See Fig. 2 (*h*). The pitch of the rivets in the inner rows is $3\frac{3}{4}$ inches; in the outer row, $7\frac{1}{2}$ inches. The rivet holes are

1 inch in diameter. The plate is $\frac{1}{2}$ inch thick and has an ultimate tensile strength of 55,000 pounds. Take the shearing strength of iron rivets at 38,000 pounds per square inch of section.

SOLUTION.—According to the statement in Art. **18,** a strip $7\frac{1}{2}$ inches wide is to be considered. The resistance of this strip to tensile stress, by rule **1,** is $55,000 \times \frac{1}{2} \times 7\frac{1}{2} = 206,250$ pounds. By rule **2,** the resistance of the net section of the plate is $(7\frac{1}{2} - 1) \times 55,000 \times \frac{1}{2} = 178,750$ pounds. In the strip we are considering, there are four rivets in double shear and one rivet in single shear. Then, the total shearing resistance is the sum of the resistances of the five rivets. By rule **3,** the resistance of the rivet in single shear is $1^2 \times .7854 \times 1 \times 38,000 = 29,845.2$ pounds. By the same rule, the resistance of the four rivets in double shear is $1^2 \times .7854 \times 4 \times 38,000 \times 1.85 = 220,854.5$ pounds, nearly. Hence, the total resistance to the shearing stress is $220,854.5 + 29,845.2 = 250,699.7$ pounds. The calculations show the net section of the plate to be weakest. The efficiency of the joint is

$$\frac{178750}{206250} \times 100 = 86.66 \text{ per cent. Ans.}$$

DESIGNING RIVETED JOINTS.

INTRODUCTION.

24. Limitations to the Application of Theory.—Although it may at first appear possible to design a riveted joint to give the maximum strength for a certain plate and quality of rivets merely by an application of the principles of the strength of materials, a closer investigation of the problem will prove that, owing to the number of conditions imposed, it is impossible to formulate a theoretical rule for the solution of the general problem. Either the diameter of the rivets or the pitch may be assumed and a joint designed that will fulfil the conditions of equality between the strength of rivets and the strength of plate; but it is impossible to derive a general rule by means of which the diameter or the pitch of rivets that will secure the *maximum strength of joint* for a given thickness and quality of plate can be directly calculated. On account of these limitations, the design of a joint that will give the maximum efficiency is a tentative problem, where either the diameter or the pitch may be

assumed and the efficiency of the corresponding joint calculated. Another size or pitch of rivet may then be chosen and the efficiency of the corresponding joint calculated. The second efficiency compared with the first will show in which direction a change in dimensions will be likely to give the best results. A few trials of this kind will reveal the dimensions required to give the most satisfactory results.

25. Practical considerations limit the design to a narrow range of values of diameters and pitch. Commercial sizes of rivets and thickness of plates, the greatest pitch with which a steam-tight joint can be secured, and the smallest pitch that will permit of the formation of rivet heads must all be considered. It is also customary to make the pitch some even number of inches or, when fractions are used, to restrict them to such numbers as halves, quarters, or eighths of an inch.

In consequence of the limitations within which theory can be applied and the practical features of the work, many designs of riveted joints are based on a series of simple empiric rules and on tables of dimensions that have been found by experience to give satisfactory results.

Regarding the pitch of rivets that must not be exceeded if the joint is to be steam-tight when under pressure, competent authorities differ somewhat. This difference of opinion accounts for much of the difference in results that will be obtained from the formulas proposed by different authorities on boiler design.

PRACTICAL RULES AND FORMULAS.

26. The symbols given below are used in the rules following:

p = pitch of rivets;
n = number of rows of rivets;
P_m = maximum pitch allowable;
P_d = diagonal pitch;
d = diameter of rivet hole;

d' = diameter of rivet;
t = thickness of plate;
v = distance between adjacent rows of rivets;
e = distance from center of rivet to edge of plate;
l = lap.

27. Board of Supervising Inspectors' Rules.—The Board of Supervising Inspectors of Steam Vessels, in their rules and regulations governing the construction of steam boilers for marine purposes, prescribe the following rules for single-riveted and double-riveted lap joints:

$d' = t + \frac{3}{8}$ inch for iron plates and iron rivets, single-riveted lap joints. (*a*)

$d' = t + \frac{5}{16}$ inch for iron plates and iron rivets, double-riveted lap joint. (*b*)

$d' = t + \frac{7}{16}$ inch for steel plates and steel rivets, single-riveted lap joint. (*c*)

$d' = t + \frac{3}{8}$ inch for steel plates and steel rivets, double-riveted lap joint. (*d*)

$$p = \frac{.7854\, d'^2 n}{t} + d' \text{ for iron plates and iron rivets.} \quad (e)$$

$$p = \frac{.645\, d'^2 n}{t} + d' \text{ for steel plates and steel rivets.} \quad (f)$$

$P_m = 1.31\, t + 1\frac{5}{8}$ inches for single-riveted lap joints. (*g*)

$P_m = 2.62\, t + 1\frac{5}{8}$ inches for double-riveted lap joints. (*h*)

$$P_d = \frac{6\,p + 4\,d'}{10} \text{ for double zigzag-riveted lap joints.} \quad (i)$$

$$v = \frac{4\,d' + 1}{2} \text{ for double chain-riveted lap joints.} \quad (k)$$

$$v = \frac{\sqrt{(11\,p + 4\,d') \times (p + 4\,d')}}{10} \text{ for double zigzag-riveted lap joints.} \quad (l)$$

$e = 1.5\, d'$. (*m*)

28. American Boiler Manufacturers Association Rules.—The American Boiler Manufacturers Association, in their "Uniform American Boiler Specifications," adopted October, 1898, in section 10 specify:

Approximately make rivet holes double the thickness of thinnest plate, or

$$d = 2\,t, \text{ approximately.} \qquad (n)$$

Make pitch three times the diameter of the rivet hole, or

$$p = 3\,d. \qquad (o)$$

Make the distance between the center line of rows of staggered rivets equal to one-half the pitch, or

$$v = .5\,p. \qquad (p)$$

Make lap for single-riveting to be equal to the pitch, or

$$l = p. \qquad (q)$$

Make lap for double-riveting one and one-third times the pitch, or

$$l = 1\tfrac{1}{3}\,p. \qquad (r)$$

For each additional row of rivets in excess of double-riveting add one-half the pitch to the lap, or

$$l = 1\tfrac{5}{6}\,p, \text{ for triple-riveting.} \qquad (s)$$
$$l = 2\tfrac{1}{3}\,p, \text{ for quadruple-riveting.} \qquad (t)$$

Exact dimensions of the joint to be determined by making the resistance to shear of rivets at least 10 per cent. greater than the resistance to tensile stress of the net section of the plate.*

29. English Board of Trade Rules.—For double-riveted butt joints with two cover-plates, the English Board of Trade prescribes that the pitch of the rivets must not exceed that given by the following formula:

$$P_m = 3.5\,t + 1\tfrac{5}{8} \text{ inches.} \qquad (u)$$

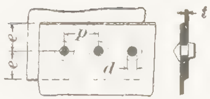
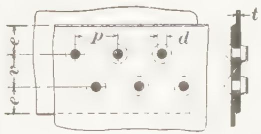
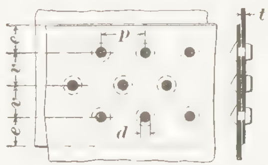
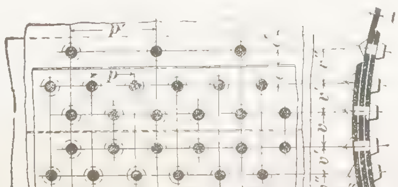
For triple-riveted butt joints with two cover-plates, the same authority gives

$$P_m = 4.63\,t + 1\tfrac{5}{8} \text{ inches.} \qquad (v)$$

* Mr. Barnet Le Van, Trenton, N. J., a high authority on this subject, states that he prefers to make the resistance to tensile stress of the net section of the plate in a new boiler larger than the resistance to shear of the rivets, since corrosion will weaken the plate, while scarcely affecting the rivets.

TABLE I.

PROPORTIONS OF RIVETED JOINTS.

Joint.	Position of Joint.	t	t	t	d	d	l	[illegible]	[illegible]	[illegible]	[illegible]	p	p	Ultimate Tensile Strength.	Ultimate Shearing Strength.	Efficiency. Per Cent.
	Girth seam, when longitudinal seams are double-riveted	1/4			11/16	3/4	1 1/8					2 1/16				
		5/16			3/4	13/16	1 7/32					2 1/8				
		3/8			7/8	15/16	1 13/32					2 3/8				
		7/16			15/16	1	1 1/2					2 [illegible]				
		1/2			1	1 1/16	1 19/32					2 [illegible]				
	Girth seam, when longitudinal seams are triple riveted.	1/4			5/8	11/16	1 1/32					2 1/16				
		5/16			11/16	3/4	1 1/8					2 1/8				
		3/8			3/4	13/16	1 7/32					2 1/8				
		7/16			7/8	15/16	1 13/32					2 3/8				
		1/2			15/16	1	1 1/2					2 [illegible]				
	Longitudinal seam.	1/4			11/16	3/4	1 1/8			1 11/16		2 7/8		60,000	38,000	74.0
		5/16			3/4	13/16	1 7/32			1 11/16		2 7/8		60,000	38,000	72.0
		3/8			7/8	15/16	1 13/32			2 3/16		3 1/4		60,000	38,000	71.0
		7/16			15/16	1	1 1/2			2 3/16		3 1/4		60,000	38,000	70.0
		1/2			1	1 1/16	1 19/32			2 5/16		3 [illegible]		60,000	38,000	70.0
	Longitudinal seam.	1/4			5/8	11/16	1 1/32			2		3		60,000	38,000	77.0
		5/16			11/16	3/4	1 1/8			2 1/16		3 1/8		60,000	38,000	76.0
		3/8			3/4	13/16	1 7/32			2 3/16		3 1/4		60,000	38,000	75.0
		7/16			7/8	15/16	1 13/32			2 [illegible]		3 3/4		60,000	38,000	75.0
		1/2			15/16	1	1 1/2			2 [illegible]		3 15/16		60,000	38,000	75.0
	Longitudinal seam.	5/16	1/4	1/4	11/16	3/4	1 1/4	1 1/4	2 [illegible]	2 [illegible]	2 1/8	3 [illegible]	6 1/4	55,000	38,000	88.0
		3/8	5/16	5/16	3/4	13/16	1 [illegible]	1 [illegible]	2 [illegible]	2 [illegible]	2 3/16	3 [illegible]	6 1/2	55,000	38,000	87.5
		7/16	3/8	3/8	7/8	15/16	1 13/32	1 13/32	2 3/4	2 13/16	2 [illegible]	3 [illegible]	6 3/4	55,000	38,000	86.0
		1/2	7/16	7/16	15/16	1	1 1/2	1 1/2	3	3	2 [illegible]	3 [illegible]	7 1/2	55,000	38,000	86.6
		9/16	7/16	7/16	1	1 1/16	1 19/32	1 19/32	3 3/16	3 3/16	2 [illegible]	3 [illegible]	7 3/4	55,000	38,000	85.4
		[illegible]	[illegible]	[illegible]	1	1 1/16	1 19/32	1 19/32	3 [illegible]	3 [illegible]	2 [illegible]	3 [illegible]	7 [illegible]	55,000	38,000	84.0

For finding the diagonal pitch for butt joints with one or two cover-plates formula (*i*) may be used.

When one cover-plate is used, make its thickness not less than $1\frac{1}{8}\,t$. (*w*)

For two cover-plates, the thickness of each should not be less than $\frac{5}{8}\,t$. (*x*)

30. Hartford Steam Boiler Inspection and Insurance Company's Proportions.—The Hartford Steam Boiler Inspection and Insurance Company have designed and recommend the joints given in Table I, the designs being for steel plate and iron rivets. With the values of the ultimate tensile and shearing strengths given, the efficiency of these joints is found in the last column.

DESIGNING A JOINT FOR THE HIGHEST PRACTICABLE EFFICIENCY.

31. The efficiency of a riveted joint depends on several interdependent factors: the diameter of the rivet when driven, the pitch of the rivets, the ultimate tensile and shearing strength of the material, and the disposition of the rivets. A change in either one or more of these factors will alter the efficiency of the joint.

For illustration, take a single-riveted lap joint for a plate $\frac{5}{16}$ inch thick, having a diameter of rivet hole of $\frac{13}{16}$ inch and a pitch of $2\frac{1}{8}$ inches. The plate having a tensile strength of 60,000 pounds and the rivet a shearing strength of 38,000 pounds, this joint will have an efficiency of 49.5 per cent., nearly.

All other data remaining the same, with the exception of the tensile strength of the plate, which we will now take at 55,000 pounds, the same joint will have an efficiency of 53.9 per cent., nearly.

The student is here cautioned against the error of assuming that a low tensile strength of the plate will be conducive to a stronger joint. The calculations here given are merely intended to show that a change in one or more of the factors

entering into the problem will work a corresponding change in the ratio between the strength of the joint and the strength of the solid plate.

When calculating the efficiency, it will be seen that in either case the joints just considered will fail by shearing the rivets. Hence, it can be made stronger by making the rivets larger.

32. Order of Procedure.—The general order of procedure outlined below may be followed in designing a joint:

1. Fix on the maximum permissible pitch for the type of joint and the thickness of plates to be used.

2. Choose a standard diameter of rivet that agrees as closely to practice for this style of joint as may be determined by the data available; in the absence of data use judgment.

3. Calculate the shearing strength of the rivet or rivets in the strip whose width is determined by the pitch, and the width of a strip of the plate that will have a tensile strength equal to the shearing strength of the rivet, or rivets.

4. Add the width of this strip to the diameter of the rivet hole, and thus obtain the pitch of the rivets required to make the tensile strength of the plate equal to the shearing strength of the assumed rivets.

5. Compare the pitch thus obtained with the assumed maximum permissible pitch; if the calculated pitch is greater than the permissible pitch, calculate again, using a smaller rivet; if it is less, try a larger rivet.

6. In this way determine the sizes of two rivets that will give pitches nearest the maximum permissible pitch, one greater and one less.

7. Calculate the efficiency of the joint whose pitch of rivets is next lower than the maximum permissible pitch.

8. Calculate the efficiency of a joint having the maximum permissible pitch, using the larger of the two rivets found in paragraph 6.

9. Of these two joints, the one that has the higher efficiency will be the joint having the highest efficiency obtainable with commercial sizes of rivets, under the assumed conditions.

This order of procedure may be considerably varied to suit special conditions. In many cases a simple inspection will enable an experienced designer to determine most of the features of the joint without the necessity of making all the calculations involved in the above outline. For the beginner, however, it is best to check each step by careful calculation.

33. Illustrative Example.—Design a double zigzag-riveted butt joint with two cover-plates, the shell plate being steel $\frac{3}{4}$ inch thick and having a tensile strength of 60,000 pounds per square inch of section. The rivets are of steel having a shearing strength of 46,000 pounds per square inch of section.

Referring to the rules for proportions of joints, we find that the only rule for fixing the limit of pitch for the kind of joints under consideration is that prescribed by the English Board of Trade [see formula (u)]; in accordance with this formula, the maximum permissible pitch for our joint is $P_m = 3.5 \times \frac{3}{4} + 1\frac{5}{8} = 4.25$ inches.

There are no rules given for the diameter of rivets for a double-riveted butt joint, but by referring to the proportions recommended by the Hartford Steam Boiler Inspection and Insurance Company, we find that a triple-riveted butt joint in a $\frac{5}{8}$-inch plate is provided with 1-inch rivets that have a shearing strength of 38,000 pounds per square inch; although these conditions differ somewhat from the ones we are considering, they may be taken as an indication of the most probable size that we shall need, and we will, therefore, use a 1-inch rivet for our first *trial size*. For this size, the shearing strength of the two rivets included within a strip of plate having a width corresponding to the pitch is, by rule **3**, $R_s = (1\frac{1}{16})^2 \times .7854 \times 2 \times 46{,}000 \times 1.85 = 150{,}900$ pounds. It will be observed that in this calculation the diameter of

the rivet hole is used, which is $\frac{1}{16}$ inch larger than the rivet.

We will now calculate the pitch that would be required to secure the same strength for the plate. The tensile strength of a strip of the plate 1 inch in width is, by rule **1**, $1 \times \frac{3}{4} \times 60{,}000 = 45{,}000$ pounds; the net width of plate required to secure a strength equal to the shearing strength of the rivets is, therefore, $150{,}900 \div 45{,}000 = 3.353 = 3\frac{3}{8}$ inches, nearly. Since the diameter of the rivet hole is $1\frac{1}{16}$ inches, the pitch required will be $3\frac{3}{8} + 1\frac{1}{16} = 4\frac{7}{16}$ inches, which is greater than the permissible maximum pitch that we have assumed for our joint. We will now try the next smaller size of rivet, $\frac{15}{16}$ inch, making the size of the rivet hole 1 inch. The shearing strength of the rivets now becomes $R_s = 1^2 \times .7854 \times 2 \times 46{,}000 \times 1.85 = 133{,}700$ pounds, nearly, and the net width of plate required to furnish the same strength is $133{,}700 \div 45{,}000 = 2.971$ inches, or 3 inches, nearly. This corresponds to a pitch of $3 + 1 = 4$ inches, which is less than the assumed permissible maximum.

34. The question now arises: *Will the efficiency be greater with this proportion than it would be if we used the next larger size of rivet and reduced the net section of the plate to a value that would give the greatest permissible pitch?* With the permissible pitch of $4\frac{1}{4}$ inches, the strength of the solid plate is $45{,}000 \times 4\frac{1}{4} = 191{,}250$, say 191,300 pounds. With a 1-inch rivet, the strength of the net section will be $(4\frac{1}{4} - 1\frac{1}{16}) \times 45{,}000 = 143{,}400$ pounds and the efficiency will be $\frac{143{,}400 \times 100}{191{,}300} = 74.96$ per cent. With the $\frac{15}{16}$-inch rivet and the pitch of 4 inches, the strength of the solid plate is $4 \times 45{,}000 = 180{,}000$ pounds. The strength of the joint is the same as that of the rivets; the efficiency is, therefore, $\frac{133{,}700 \times 100}{180{,}000} = 74.28$ per cent., very nearly. This is less than the efficiency with the next larger size of rivets and the maximum permissible pitch, and a little inspection shows that any smaller size of rivets would give a still lower

efficiency. Also, since the 1-inch rivets are stronger than the net section of the plate with a pitch of 4 inches, we see that any attempt to use larger rivets with this pitch would only result in further reducing the strength of the net section and, in consequence, the efficiency.

If the pitch could be increased to $4\frac{7}{16}$ inches, so as to make the strength of the net section of the plate equal to that of the 1-inch rivet, the strength of the solid plate would be $45,000 \times 4\frac{7}{16} = 199,700$ pounds and the efficiency would be increased to $\frac{150,900 \times 100}{199,700} = 75.56$ per cent. This, however, is contrary to the rule that was available for the greatest pitch allowable. We therefore see that the maximum efficiency that can be obtained *under the assumed conditions* is 74.96 per cent.

ATTAINABLE EFFICIENCIES.

35. The efficiencies of riveted joints that may be expected to be attained in practice average as follows:

For a single-riveted lap joint, 56 per cent.

For a double-riveted lap joint, 70 per cent.

For a triple-riveted lap joint, 75 per cent.

For a double-riveted butt joint with two cover-plates, 76 per cent.

For a triple-riveted butt joint with two cover-plates, 85 per cent.

Single-riveted and double-riveted butt joints with one cover-plate give about the same efficiencies as single-riveted and double-riveted lap joints.

The efficiencies given must be understood to be nothing else but *average* efficiencies; the student must not fall into the error of assuming that because a lap joint is triple-riveted it *must* have an efficiency of 75 per cent. As a matter of fact, it may be slightly higher and can be much lower. Hence, the efficiency should always be calculated by the rules given.

Numerous tests have shown the average shearing resistance of iron and steel rivets in single shear and per square inch of section to average, for iron rivets, 38,000 pounds; for steel rivets, 46,000 pounds.

THE ARRANGEMENT AND DESIGN OF STAYS.

ARRANGEMENT.

36. Distribution of Stays.—In staying boilers, it should be the aim of the designer to distribute the stays as equably as possible over the area to be braced. The process of laying out the bracing is a tentative one, involving the use of judgment and common sense. After the number of braces of a given size has been determined by calculation, it will frequently be found that with the chosen number of braces an equable distribution is out of the question. Then a different number of braces, and, perhaps, even a different size of brace, will have to be chosen.

The arrangement of braces depends, to a large extent, on the shape of the area to be braced. For rectangular areas, such as are presented by the firebox sheets of boilers of the locomotive type, the most common and natural method is to arrange the stays in equidistant horizontal and vertical rows. For bracing the heads of return-tubular boilers, the Hartford Steam Boiler Inspection and Insurance Company advocates arranging the braces in concentric rows,

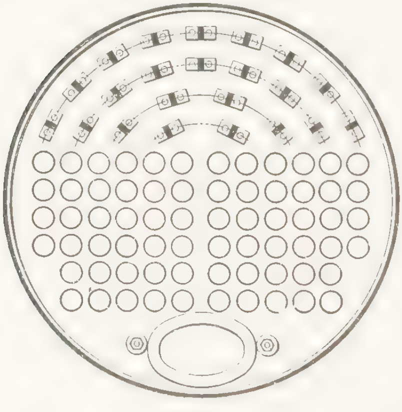


Fig. 4

as shown in Fig. 4, the design shown being for a 72-inch

boiler braced by crowfoot braces riveted to the head and shell. When the head is stiffened by T irons, an excellent arrangement of them is that shown in Fig. 5. As will be observed, the T irons are arranged radially, and each is securely riveted to the shell. The forked end of the braces is then cottered to the T iron between the rivets, as shown. The great advantage of this system of bracing is that by it the number of braces is reduced, thus making the boiler more accessible for cleaning and repairing. The reduction in the number of braces naturally requires each brace to be larger.

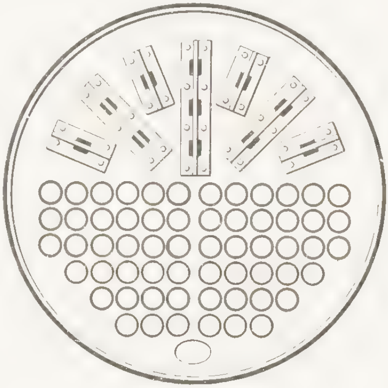


FIG. 5.

37. The area supported by each stay or brace is the area bounded by lines midway between the stay and the nearest stays surrounding it. Thus, in Fig. 6, the dotted lines show the area supported by the central stay. When stays are arranged in equidistant rows, as shown in the figure, which is the usual way of arranging them in the flat surfaces of firebox boilers, the area supported is equal to the square of the center-to-center distance, or *pitch*, of the stays.

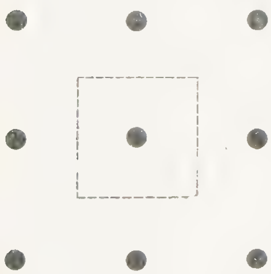


FIG. 6.

38. Area to be Supported.—In calculating the total area that is to be supported, it can safely be assumed, for a boiler head, that the flanging of the head will support it for a distance equal to 6 times the thickness of the plate. Likewise, with the flat surfaces of the firebox, the flanging of the end sheets will support them to a distance that may be

taken equal to 6 times the thickness of the plate. In the case of the head of a horizontal return-tubular boiler, the tubes, when well rolled, act as stays and render further staying of this part of the boiler unnecessary. The tubes may also be assumed to safely support the head for a distance equal to about $1\frac{1}{4}$ times the diameter of the tube, measuring from the center of the upper row of tubes. Thus, in Fig. 7, with a boiler head $\frac{1}{2}$ inch thick and tubes $3\frac{1}{2}$ inches in diameter, the areas to be supported by staying are those enclosed by dotted lines. The flange of the head supports it for $\frac{1}{2} \times 6 = 3$ inches; the tubes support the head for a distance of $3\frac{1}{2} \times 1\frac{1}{4} = 4\frac{3}{8}$ inches from the center of the tubes. The total area to be supported can now be calculated by the rules of mensuration, taking the necessary measurements from a scale drawing. The total pressure, or load, on the area to be supported is the product of the steam pressure and the area in square inches.

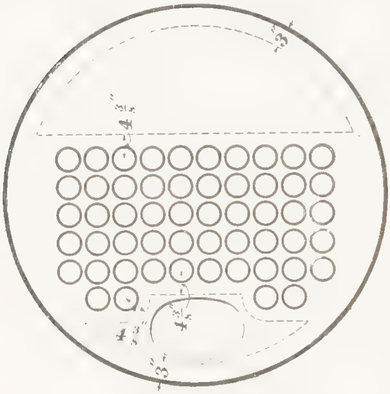


FIG. 7.

RULES FOR DESIGNING STAYS.

39. Symbols.—

Let t = thickness of plate in sixteenths of an inch;
P = steam pressure in pounds per square inch;
A = area supported by one stay in square inches;
l = length of center line of diagonal stay, measured between head and shell, in inches;
b = distance from intersection of center line of diagonal stay with shell to the inside of the boiler head, measured parallel to the shell, in inches;

L = load on one stay;
S = load per square inch of section of the stay;
a = area of stay in square inches.

40. Maximum Permissible Distance Between Stays. Considering the thickness of the plate to be braced, it is evidently necessary to have the braces or stays close enough to each other to prevent excessive bulging between the stays. In other words, the area that each stay or brace supports should not exceed a certain number of square inches, the area depending on the thickness of the plate and the steam pressure to be carried. This area has been determined experimentally and is given by the following rule, which has been deduced from rules of the Board of Supervising Inspectors of Steam Vessels, and is also prescribed in the specifications of the American Boiler Manufacturers Association.

Rule 4.—*Multiply 112 by the square of the thickness of the plate in sixteenths of an inch, and divide the product by the steam pressure in pounds per square inch. The quotient will be the maximum area, in square inches, that can be supported safely by one stay. For plates above $\frac{7}{16}$ inch in thickness, use 120 instead of 112. When screw staybolts are fitted with nuts inside and outside of each plate, use 140 as a constant.*

Or, $A = \frac{112\,t^2}{P}$ for plates up to $\frac{7}{16}$ inch thick,

and $A = \frac{120\,t^2}{P}$ for plates above $\frac{7}{16}$ inch thick,

and $A = \frac{140\,t^2}{P}$ for screw staybolts with nuts.

From this rule, the maximum pitch of stays arranged in equidistant rows can readily be found by extracting the square root of the area given by rule 4.

The rules and regulations of the Board of Supervising Inspectors of Steam Vessels provide that braces or staybolts must not be placed more than $10\frac{1}{2}$ inches from center to center when used to brace the flat surfaces of fireboxes, furnaces, and back connections.

EXAMPLE.—With a $\frac{1}{2}$-inch plate to carry a steam pressure of 125 pounds, what is (*a*) the greatest area that one stay can support when nuts are not fitted to the stays? (*b*) The stays being arranged in equidistant rows, what is the maximum pitch?

SOLUTION.— $\frac{1}{2}$ inch $= \frac{8}{16}$. Then, applying rule **4**, we get

$$(a)\ A = \frac{120 \times 8^2}{125} = 61.44 \text{ sq. in.} \quad \text{Ans.}$$

$$(b) \qquad \sqrt{61.44} = 7.83 \text{ in.} \quad \text{Ans.}$$

41. The preceding rule may advantageously be applied to the heads and other flat surfaces of stationary boilers as a test of the spacing of the braces and stays. For example, suppose that it has been decided to use 8 stays on the upper part of the 60-inch boiler head shown in Fig. 7. Let the area be 720 square inches. Then, with the stays equably distributed, each stay will support $720 \div 8 = 90$ square inches. Suppose the head is $\frac{7}{16}$ inch thick and the steam pressure is to be 100 pounds. Then, by rule **4**, the maximum area is $\frac{112 \times 7^2}{100} = 54.88$ square inches. This shows that there are not enough braces for the given steam pressure and thickness of boiler head. Hence, if the steam pressure cannot be lowered or the thickness of the head increased, the number of braces must be increased.

Knowing the area allowable for each stay and the total area to be supported, the minimum number of stays is evidently the total area divided by the area allowable for each stay. Thus, taking the same example as above, there should be not less than $720 \div 54.88 = 13+$, say 14 braces. When plotting the distribution of braces over irregular surfaces, it will often be found that, with the number of braces calculated as above, an equable distribution is impossible. In that case use the next larger number that will insure a good distribution. For instance, in a design of head bracing for a 60-inch boiler head to carry 100 pounds pressure, designed and advocated by the Hartford Steam Boiler Inspection and Insurance Company, we find 15 braces used against 14 found by calculation.

42. Load on Stay.—Having determined the number of braces or stays to be used, the next step is to find the load each stay has to support. This load, in the case of direct stays (stays at a right angle to the surface), is equal to the product of the area supported and the steam pressure. Thus, for stays in equidistant rows and with a 6-inch pitch, for a steam pressure of 100 pounds, the load on each stay will be $6^2 \times 100 = 3,600$ pounds.

In the case of crowfoot and other stays making an angle with the surface supported, the load on each stay will be increased by reason of this angle. The load on such a stay can be found by the following rule:

Rule 5.—*Divide the length of the center line of the stay, measured between the points of intersection of this center line with the head and shell, by the perpendicular distance between the intersection of the center line of the stay with the shell and the head. Multiply the quotient thus obtained by the product of the area and the steam pressure.*

Or, $$L = \frac{l}{b} A P.$$

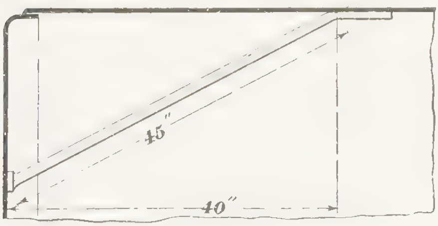


FIG. 8.

EXAMPLE.—In the crowfoot stay shown in Fig. 8, what is the load on the stay when it supports an area of 60 square inches and the steam pressure is 100 pounds?

SOLUTION.—Applying the rule just given, we have

$$L = \frac{45}{40} \times 60 \times 100 = 6,750 \text{ lb. Ans.}$$

With stays making an angle with the surface that they are to support, the load on each stay should be calculated, as the different stays make different angles with the surface.

43. Sectional Area of Stay.—Having determined the load on the stays or braces, the minimum area of the stays or braces that is sufficient to safely bear this load must be found. In all boilers constructed for marine work, the

greatest load per square inch of section of the brace is fixed by law at 6,000 pounds. In stationary boilers, it is usual to allow a load as high as 7,500 pounds per square inch of section. However, the specifications of the American Boiler Manufacturers Association provide that when the material for braces has been tested and found to conform to their specifications, iron braces may be allowed a load of 9,000 pounds per square inch and steel braces a load of 11,000 pounds per square inch. When no test of the material has been made, the load must not exceed 6,500 pounds per square inch for wrought iron and 8,000 pounds per square inch for steel. Then, to find the minimum area of the brace or stay, that is, the area at the smallest part, expressed in square inches, we have the following rule:

Rule 6.—*Divide the load on the stay by the allowable load per square inch of section.*

Or,

$$a = \frac{L}{S}.$$

Example.—What must be the area of a crowfoot stay on which the load is 7,000 pounds, allowing a load of 7,500 pounds per square inch of section?

Solution.—Applying rule **6**, we get

$$a = \frac{7000}{7500} = .933 \text{ sq. in.}$$

The corresponding diameter is $1\frac{1}{8}$ in., nearly. Ans.

44. Application of Rules.—With the rules here given, all problems relating to the staying of boilers by direct and diagonal stays may be solved. The practical application of these rules will now be shown by some regular questions asked in engineers' examinations.

Example 1.—How many $\frac{7}{8}$-inch staybolts are required for a crown sheet of a locomotive boiler to carry 120 pounds steam pressure? The sheet is 4 ft. × 6 ft. Allow a load of 6,000 pounds per square inch of section on the stay.

Solution.—In this example the thickness of the plate is not given, hence the number of staybolts cannot be calculated from rule **4**. But, since the size of the staybolts is known, the load each staybolt can bear can be readily calculated, and by dividing the total load by the load on

each staybolt, the number of staybolts can be obtained. The area of the sheet, in square inches, is $4 \times 12 \times 6 \times 12 = 3,456$ square inches. The total load is $3,456 \times 120 = 414,720$ pounds. The area of a $\frac{7}{8}$-inch staybolt, assuming that the bolt is $\frac{7}{8}$ inch at the bottom of the thread, is $(\frac{7}{8})^2 \times .7854 = .601$ square inch, and the load it can bear is $.601 \times 6,000 = 3,606$ pounds. Then, $414,720 \div 3,606 = 115$ staybolts. Ans.

EXAMPLE 2.—There are 48 staybolts in a sheet 40 in. $\times$ 50 in. What size should they be for a steam pressure of 100 pounds?

SOLUTION.—The total load on the sheet is $40 \times 50 \times 100 = 200,000$ pounds. Since there are to be 48 staybolts, the load each staybolt will bear when equably distributed is $200,000 \div 48 = 4,167$ pounds, nearly. Allowing a load of 6,000 pounds per square inch of section, the minimum area, by rule **6**, will be $4,167 \div 6,000 = .6945$ square inch. The diameter corresponding to this is $\sqrt{\frac{.6945}{.7854}} = \frac{15}{16}$ in., nearly. Ans.

45. Since the thickness of the plate is not mentioned in the two examples just given, it is impossible to say whether or not there will be excessive bulging of the plate between the stays when the pressure is applied. The questions given are actual examination questions that have been asked candidates for a stationary engineer's license. The answers given and the method of working the problems, while usually satisfactory to the examiner, are scarcely satisfactory to a designer. For an actual design it is recommended that these calculations be checked by considering the thickness of the plate and applying rule **4**, thus preventing an excessive center-to-center distance of the stays.

RIVETS FOR BRACING.

46. Evidently the rivets securing the braces of the crowfoot type to the head and shell should, at least, be as strong as the brace itself. The rivets securing the crowfoot to the head are not in direct tension and cannot be calculated for a straight pull, as can be done for the brace itself. Generally speaking, it has been found that if the combined sectional area of the two rivets in the crowfoot is made equal to $1\frac{1}{4}$ times the minimum area of the brace, they will be as strong as the brace. Thus, for a crowfoot

brace $1\frac{1}{4}$ inches in diameter, the area of which is $(1\frac{1}{4})^2 \times .7854 = 1.227$ square inches, the combined area of the rivets should be $1.227 \times 1.25 = 1.524$ inches, whence the area of each rivet is $1.524 \div 2 = .762$ square inch. The corresponding diameter is $\sqrt{\frac{.762}{.7854}} =$ say, 1 inch.

A great many boilermakers make the area of each rivet equal to one-half the area of the brace, but, as this will leave the rivets weaker than the brace, it is better to make them larger.

47. The rivets securing the shank of crowfoot and similar braces are in single shear and may be calculated from the shearing strength of the material. In practice, however, it is the usual custom to make their combined area equal to $1\frac{1}{4}$ times the area of the stay. The rivets securing **T** and angle irons to heads are in direct tension and may be calculated for tension, taking a low value for the safe tensile strength. In general, it will not be advisable to allow a greater load than 4,000 pounds per square inch of section on the rivet. Rivets smaller than $\frac{3}{4}$ inch should never be used.

Having laid out the rivet centers on the drawing board, spacing them as equably as feasible, the combined cross-sectional area of all the rivets can readily be found. Thus, if the area to be supported is 720 square inches and the pressure 100 pounds, the total load is $720 \times 100 = 72,000$ pounds. If there are 31 rivets, each rivet must bear a load of $72,000 \div 31 = 2,323$ pounds, nearly. Then, the area of each rivet is $2,323 \div 4,000 = .58$ square inch. The corresponding diameter is $\sqrt{\frac{.58}{.7854}} =$ say, $\frac{7}{8}$ inch.

48. In laying out the rivet centers it is not always possible to distribute them evenly over the surface to be braced. A case in point is shown in Fig. 5. As will be observed, the rivets have been arranged in groups of two, and these groups are distributed as evenly as possible. In such a

case, it is well to calculate the rivet size for the assembly of groups having the least number of rivets for the largest number of stays. Thus, in Fig. 5, the vertical **T** iron in the center carries 3 stays with 8 rivets. Plot on the drawing board the area supported by the 3 stays; measure it, and then multiply by the steam pressure to get the total load on the 8 rivets. From this obtain the diameter of the rivet in the way previously explained. It is the usual practice to use the same size of rivets for all head bracing; hence, the rivets that carry a smaller load should be made equal to those carrying a larger load.

UNSTAYED HEADS.

49. Boiler heads not supported by staying are only found in plain cylindrical boilers. Unstayed heads are also used in the construction of steam drums and mud drums. They are usually made in one of two forms, as shown in Fig. 9. When fitted so that the concave or hollow side receives the steam pressure, as shown in Fig. 9 (*a*), they are called **bumped heads**; when the convex side receives the steam pressure, as in Fig. 9 (*b*), they are termed **concaved heads.** Occasionally the heads are made flat.

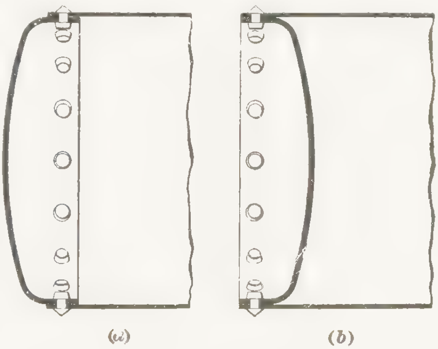


Fig. 9.

Let P = working pressure in pounds per square inch;
T = thickness of head in inches;
R = radius in inches to which the head is bumped;
S = ultimate tensile strength of the material;
A = area of head in square inches.

Then, the working pressure may be obtained by the rules below. When the head is bumped and single-riveted to the shell:

Rule 7.—*Multiply the thickness of the plate by its tensile strength and divide by 3 times the radius to which the head is bumped.*

Or, $$P = \frac{T\,S}{3\,R}.$$

EXAMPLE.—What working pressure would you allow on a bumped head, bumped to a radius of 72 inches, when the head is ½-inch thick and made of material having a tensile strength of 55,000 pounds? The head is single-riveted.

SOLUTION.—Applying the rule just given, we have

$$P = \frac{\frac{1}{2} \times 55{,}000}{3 \times 72} = 127.3 \text{ lb. Ans.}$$

50. When the head is double-riveted to the shell:

Rule 8.—*Multiply the thickness of the plate by its tensile strength and divide by 2.5 times the radius to which the head is bumped.*

Or, $$P = \frac{T\,S}{2.5\,R}.$$

EXAMPLE.—In the last example, what working pressure would you allow if the head is double-riveted?

SOLUTION.—Applying rule **8**, we get

$$P = \frac{\frac{1}{2} \times 55{,}000}{2.5 \times 72} = 152.8 \text{ lb. Ans.}$$

51. When the head is concaved and single-riveted to the shell:

Rule 9.—*Multiply the thickness of the plate by its tensile strength and divide by 5 times the radius to which the head is bumped.*

Or, $$P = \frac{T\,S}{5\,R}.$$

EXAMPLE.—Taking the same example as before, what working pressure would be allowable if the head was concaved instead of bumped?

SOLUTION.—Applying the rule just given, we get

$$P = \frac{\frac{1}{2} \times 55,000}{5 \times 72} = 76.4 \text{ lb. Ans.}$$

52. When a concaved head is double-riveted to the shell:

Rule 10.—*Multiply the thickness of the plate by its tensile strength and divide the product by $4\frac{1}{6}$ times the radius to which the head is bumped.*

Or, $$P = \frac{TS}{4\frac{1}{6} R}.$$

EXAMPLE.—Taking the last example, what pressure would be allowed if the head was double-riveted to the shell?

SOLUTION.—Applying rule **10**, we get

$$P = \frac{\frac{1}{2} \times 55,000}{4\frac{1}{6} \times 72} = 91.67 \text{ lb. Ans.}$$

53. When the head is flat without any staying:

Rule 11.—*Multiply the thickness of the head by its tensile strength and divide the product by .54 times the area of the head in square inches.*

Or, $$P = \frac{TS}{.54 A}.$$

EXAMPLE.—What working pressure would you allow on a flat head 32 inches in diameter, $\frac{3}{4}$ inch thick, and having an ultimate tensile strength of 60,000 pounds?

SOLUTION.—Applying rule **11**, we get

$$P = \frac{\frac{3}{4} \times 60,000}{.54 \times 32^2 \times .7854} = 103.6 \text{ lb. Ans.}$$

Rules **7**, **8**, **9**, **10**, and **11** are those prescribed by the Board of Supervising Inspectors of Steam Vessels for the guidance of their inspectors in determining the working pressures allowable on the kinds of heads given. An examination of these rules shows that it is most economical to use the bumped head, since, with a given thickness and tensile strength, it will be allowed a greater working pressure than any other. Conversely, for a given working pressure, a thinner head may be used.

TABLE II.

STANDARD SIZES OF BOILER TUBES.

Diameter. Inches.	Thickness. Inches.	Wire Gauge Number.	Transverse Internal Area. Square Inches.	Length of Tube per Square Foot of Internal Surface. Feet.
1	.072	15	.575	4.462
$1\frac{1}{4}$	.072	15	.961	3.453
$1\frac{1}{2}$	.083	14	1.398	2.863
$1\frac{3}{4}$	.095	13	1.911	2.448
2	.095	13	2.573	2.110
$2\frac{1}{4}$	.095	13	3.333	1.854
$2\frac{1}{2}$	.109	12	4.090	1.674
$2\frac{3}{4}$	.109	12	5.035	1.509
3	.109	12	6.079	1.373
$3\frac{1}{4}$	.120	11	7.116	1.260
$3\frac{1}{2}$	.120	11	8.347	1.172
$3\frac{3}{4}$	.120	11	9.676	1.088
4	.134	10	10.939	1.024
$4\frac{1}{2}$	.134	10	14.066	.902
5	.148	9	17.379	.812
6	.165	8	25.249	.673
7	.165	8	34.942	.573
8	.165	8	46.204	.498
9	.180	7	58.629	.442
10	.203	6	72.292	.398
11	.220	5	87.583	.362
12	.229	$4\frac{1}{2}$	104.629	.330
13	.238	4	123.190	.305
14	.248	$3\frac{1}{2}$	143.224	.283
15	.259	3	164.720	.264
16	.284	2	187.040	.248

TUBES AND FLUES.

STANDARD SIZES.

54. Tubes used for increasing the heating surface of boilers are made of charcoal iron or soft steel and are lap-welded or solid drawn. When tubes exceed 6 inches in external diameter, they are commonly spoken of as **flues.** Unlike pipes, the size of tubes for boiler work is designated by their external diameter, which is the *actual* diameter and not the nominal diameter, as in case of pipes used for conveying fluids. The standard dimensions of the tubes most commonly used are given in Table II.

ALLOWABLE EXTERNAL PRESSURES ON TUBES.

55. Boiler tubes up to and including 6 inches in external diameter may be allowed a working pressure of 225 pounds per square inch, if made of the thickness given in the table. Flues above 6 inches in diameter, up to and including 16 inches, may be allowed a working pressure of 60 pounds per square inch, if their length does not exceed 18 feet and their thickness is as given in the table. When flues above 6 inches and not over 16 inches are made in sections not over 5 feet in length and securely riveted together, a working pressure of 120 pounds per square inch may be allowed.

Boiler tubes made of charcoal iron and lap-welded may be obtained in sizes up to and including 4 inches made one gauge thicker than those given in the table. These tubes are made especially for locomotive work and may be allowed a pressure not over 300 pounds per square inch.

ALLOWABLE PRESSURES ON FLUES.

56. The working pressure allowable on plain flues made in sections not over 8 feet in length, with the ends of each section flanged and riveted with a reenforcing ring between the flanges, may be obtained from the following rule,

where P = steam pressure allowable;
T = thickness of flue in inches;
L = length of section in feet;
D = diameter of flue in inches.

Rule 12.—*Multiply 89,600 by the square of the thickness of the flue and divide the product by the product of the length of the section and the diameter of the flue.*

Or,
$$P = \frac{89{,}600\ T^2}{L\ D}.$$

EXAMPLE.—Given, a flue made of material .3 inch thick and 20 inches in diameter. If made in sections 5 feet long, what working pressure per square inch may be allowed?

SOLUTION.—Applying the rule just given, we get

$$P = \frac{89{,}600 \times .3^2}{5 \times 20} = 80.64 \text{ lb.} \quad \text{Ans.}$$

GRATE AREA, HEATING SURFACE, AND TUBE AREA.

GRATE AREA.

57. Considerations Affecting the Grate Area.—The grate area, or grate surface, depends on the rate of combustion, the quantity of water evaporated per pound of fuel, and the total weight of steam evaporated per hour.

For example, suppose that a plant needs 8,000 pounds of steam per hour. Assume that 16 pounds of coal are burned per square foot of grate surface per hour, and that each pound of coal will evaporate 8 pounds of water. Then, it follows that the grate surface required is $\frac{8{,}000}{16 \times 8} = 62.5$ square feet. This grate surface would necessarily be divided among several furnaces, since a grate longer and wider than 6 feet cannot readily be fired.

58. Let G = area of grate in square feet;
F = rate of combustion in pounds per square foot of grate area per hour;
W = weight of steam per hour;
E = evaporation in pounds of water per pound of coal.

To find the grate surface required for a plant:

Rule 13.—*Divide the weight of steam per hour by the product of the rate of combustion and the evaporation.*

Or,
$$G = \frac{W}{FE}$$

In rule **13,** no account has been taken of the difference in the number of heat units required to evaporate water from different feedwater temperatures into steam at different pressures. Hence, the rule is only approximate, but close enough for practical work.

59. The *average* evaporation per pound of coal for different kinds of boilers is given in the following table:

TABLE III.

AVERAGE EVAPORATION PER POUND OF COAL.

Coal per Hour per Square Foot of Grate Area.	6–10	10–14	14–18	18–20
Cylinder boiler	7.00	6.75	6.50	6.00
Two-flue boiler	7.25	7.00	6.75	6.25
Return-tubular boiler	9.00	8.50	8.25	8.00
Firebox boiler	9.00	8.50	8.25	8.00
Vertical tubular boiler	8.00	7.75	7.50	7.00
Water-tube boiler	10.50	10.00	9.00	8.00

In the table, the figure below the combustion rate and on the same horizontal line as the kind of boiler gives the

evaporation per pound of coal that may be expected under average conditions. The student must not expect that the actual evaporation obtained after installing the plant will be exactly that given in the table; this table is merely intended as an approximate guide to aid the student if no data are available showing the evaporation of the kind of boiler selected under conditions similar to those under which his own plant is operated.

The number of pounds of coal that may be burned per square foot of grate surface under natural draft will be given later.

EXAMPLE.—A battery of cylinder boilers is to generate 6,000 pounds of steam per hour, burning 12 pounds of coal per square foot of grate surface per hour. What grate surface will be required?

SOLUTION.—By the table, an evaporation of 6.75 pounds of water per pound of coal may be expected. Then, applying rule **13,** we get

$$G = \frac{6{,}000}{12 \times 6.75} = 74 \text{ sq. ft., nearly.} \quad \text{Ans.}$$

HEATING SURFACE.

60. Definition.—The heating surface of a boiler is the portion of the surface exposed to the action of the flames and hot gases. This includes the portions of the shell below the line of brickwork, the exposed heads of the shell, and the interior surface of the tubes, in the case of a multitubular boiler. In the case of a water-tube boiler, the heating surface comprises the portion of the shell below the brickwork, the outer surface of headers, and outer surface of tubes.

61. To find the heating surface of a return-tubular boiler:

Rule 14.—*Multiply two-thirds the circumference of the shell in inches by its length in inches; multiply the number of tubes by the length of the tube in inches and by its circumference; add to the sum of these products two-thirds of the area in square inches of the two heads or tube-sheets; from this*

sum subtract twice the area of all the tubes and divide the remainder by 144; the result is the heating surface in square feet.

EXAMPLE.—A horizontal return-tubular boiler has the following dimensions: Diameter, 60 inches; length of tubes, 12 feet; internal diameter of tubes, 3 inches; number of tubes, 82.

SOLUTION.—

Circumference of shell	$= 60 \times 3.1416 = 188.496 = 188.5$ inches, say.
Length of shell	$= 12 \times 12 = 144$ inches.
Heating surface of shell	$= 188.5 \times 144 \times \frac{2}{3} = 18{,}096$ square inches.
Circumference of tube	$= 3 \times 3.1416 = 9.425$ inches, nearly.
Heating surface of tubes	$= 82 \times 144 \times 9.425 = 111{,}290.4$ square inches.
Area of one head	$= 60^2 \times .7854 = 2{,}827.44$ square inches.
Two-thirds area of both heads	$= \frac{2}{3} \times 2 \times 2{,}827.44 = 3{,}769.92$ square inches.
Area through tubes	$= 3^2 \times .7854 \times 82 = 579.63$ square inches.

By rule **14**,

$$\text{Heating surface} = \frac{18{,}096 + 111{,}290.4 + 3{,}769.92 - 2 \times 579.63}{144}$$
$$= 916.64 \text{ sq. ft.} \quad \text{Ans.}$$

62. To find the heating surface of a vertical tubular boiler:

Rule 15.—*Multiply the circumference of the firebox in inches by its height above the grate; multiply the number of tubes by the length of a tube in inches and by its circumference; add to the sum of these two products the area in square inches of the lower tube-sheet; from this sum subtract the area of all the tubes and divide the remainder by 144; the quotient is the desired heating surface in square feet.*

In a firebox boiler, the heating surface includes the portion of the firebox above the grate surrounded by water and the inner surface of the tubes. The heating surface of an internally fired boiler of the Cornish or Lancashire type includes the surface of the furnace flues and that part of the outer shell in contact with the gases.

63. Efficiency of Heating Surface.—The ability of a heating surface to abstract heat from the furnace or from the gases of combustion depends largely on its location in the boiler and on the character of its contact with the gases.

The very best heating surface is a flat horizontal plate above the fire, as, for example, the crown sheet of a locomotive boiler. The lower shell of a horizontal tubular boiler is not quite as efficient on account of its curvature. A vertical plate is about one-half as efficient as a horizontal plate above the fire, and a horizontal plate below the fire is nearly worthless. When the gases pass through tubes, as in tubular horizontal and vertical boilers, the tubes give up more heat to the water when horizontal than when vertical, and the first 3 or 4 feet of the tube are very much more efficient than the end near the smokebox. Further, in a horizontal tubular boiler, the water abstracts much more heat from the upper tubes than from those near the bottom of the shell. In computing the heating surface of a boiler, no account is taken of the difference of efficiency. It is a point, however, that should be carefully considered in the design of boilers.

64. Size and Arrangement of Tubes.—Since the greatest part of the heating surface of tubular boilers is furnished by the tubes, particular attention must be paid to their size and arrangement. The length of the tubes should be about 50 diameters for bituminous coal and 60 diameters for anthracite coal; these two proportions represent good modern practice. The tubes of horizontal boilers should be arranged in horizontal and vertical rows, with a horizontal spacing of from $1\frac{1}{3}$ to $1\frac{1}{2}$ times the tube diameter, preferably the latter. The vertical spacing may be somewhat less. The upper row of tubes should not be any higher than $\frac{2}{3}$ of the diameter of the boiler from the bottom in order to leave ample steam room on the top. The sizes of tubes used in ordinary practice with horizontal return-tubular boilers is as follows: for boilers between 36 and 48 inches diameter, 3-inch tubes; from 48 to 60 inches diameter, $3\frac{1}{2}$-inch tubes; from 60 to 72 inches diameter, 4-inch tubes.

65. It is common practice to divide the tubes into two nests with a large central water space, as it is thought that

such an arrangement permits the water to rise in the central space and descend on the outside of the nests next to the shell. There is little reason to doubt that the central water space will cause such circulation when the tubes are packed close together. Many authorities, however, think that much better results can be obtained by a wider and uniform horizontal spacing of tubes, since then a freer circulation can take place between each row of tubes.

A space of at least 3 inches should be left between the tubes and the shell; the bottom row of tubes should be at a sufficient distance from the bottom of the shell to allow a large body of water to rest directly on the sheets exposed to the fire. This insures good circulation and facilitates examination, cleaning, and repairs.

66. Ratio of Heating Surface to Grate Area.—In order to obtain the best results from a boiler, the temperature of the products of combustion should pass into the chimney at as low a temperature as possible. To give these hot gases a chance to give up their heat to the water, a large amount of heating surface is necessary. The higher the rate of combustion, the greater should be the heating surface.

67. In practice, the ratio between the heating surface and grate area varies with the type of boiler and the rate of combustion. The following are average values:

Type.	$\text{Ratio} = \frac{\text{Heating Surface}}{\text{Grate Area}}$.
Plain cylindrical	12 to 15
Flue	20 to 25
Multitubular	25 to 35
Vertical	25 to 30
Water tube	35 to 40
Locomotive	50 to 100

From a large number of tests, Mr. G. H. Barrus concludes that with bituminous coal a return-tubular boiler gives the best results when the ratio is between 45 to 50, provided the rate of combustion is not more than 12 pounds

per square foot of grate surface per hour. Under the same circumstances the ratio should be 36 when the boiler uses anthracite coal.

TUBE AREA.

68. Since the products of combustion must pass through the tubes or flues, their combined cross-sectional area (usually called the **tube area**) must be large enough to allow the volume of heated gases to pass through them without interfering with the draft, and be small enough to retard the flow of gases sufficiently to allow them to part with the greater part of their heat. The average practice is to make the combined cross-sectional area of the tubes or flues equal to $\frac{1}{9}$ to $\frac{1}{8}$ of the grate surface for anthracite coal, and from $\frac{1}{7}$ to $\frac{1}{6}$ of the grate surface for bituminous coal.

HORSEPOWER OF BOILERS.

69. Introduction.—Strictly speaking, there is no such a thing as the horsepower of a steam boiler. The term has come into such extensive use, however, that an explanation of its various meanings is necessary.

When first used, it signified that the boiler would furnish steam to an engine of the same horsepower; this meant that if a certain boiler furnished steam for a 30-horsepower engine, it would be called a 30-horsepower boiler; and if the same boiler furnished steam for a 50-horsepower engine, it would be called a 50-horsepower boiler. It is thus seen that this rating had no particular significance.

70. Standard Unit of Boiler Horsepower.—In order to have a definite value by which to compare boiler performances under different conditions, the American Society of Mechanical Engineers has decided that a standard boiler horsepower should be equal to the absorption of 33,330 B. T. U. by the water in the boiler. The standard boiler horsepower is given by the following rule, where W = weight of water

evaporated per hour in pounds; H = total heat of steam above 32° at pressure of actual evaporation; t = temperature of feedwater; and B = standard boiler horsepower.

Rule 16.—*Subtract the temperature of the feedwater from the total heat of 1 pound of steam above 32° at the pressure of the actual evaporation. Add 32 to the remainder and multiply this sum by the weight of water evaporated per hour. Divide the product by 33,330.*

Or, $$B = \frac{W(H - t + 32)}{33,330}.$$

EXAMPLE.—A boiler receives the feedwater at 62° and evaporates it into steam at 85 pounds gauge pressure. If 2,300 pounds of water is then evaporated per hour, what is the standard horsepower of the boiler?

SOLUTION.—The absolute steam pressure is 85 + 14.7 = 99.7, or nearly 100 pounds per square inch. At this pressure the total heat required to evaporate a pound of water from 32° is about 1,181.8 B. T. U. (See table of the Properties of Saturated Steam.) Then, applying the rule just given, we get

$$B = \frac{2,300\,(1,181.8 - 62 + 32)}{33,330} = 79.49 \text{ H. P. Ans.}$$

71. Relation Between Boiler Horsepower and Engine Horsepower.—The amount of steam used by engines per horsepower per hour varies within such wide limits that a horsepower rating based on heat absorption, that is, evaporation, is in itself no indication that a boiler of a given standard rating is the correct size for an engine of an equal power. Furthermore, the same boiler may generate widely differing quantities of steam under different conditions, the amount of steam generated depending primarily on the combustion rate and kind of fuel. Considering this fact, it is seen that the standard horsepower rating is a variable quantity of small value as a guide in the selection of a boiler.

72. Horsepower Rating Based on Heating Surface. It is the common practice of boilermakers to rate the horsepower of their boilers as a certain fraction of the heating surface expressed in square feet, each boilermaker using his

own fraction for different types of boilers. These widely differing ratios between heating surface and horsepower average about as follows:

Type.	$\text{Ratio} = \frac{\text{Sq. Ft. of Heating Surface.}}{\text{Rated Horsepower.}}$
Plain cylindrical....................	6 to 10
Flue................................	8 to 12
Multitubular........................	14 to 18
Vertical............................	15 to 20
Water tube	10 to 12

For example, a boilermaker rates his tubular boilers as having 16 square feet of heating surface to the horsepower. Then, a 35-horsepower boiler would have $35 \times 16 = 560$ square feet of heating surface. On the other hand, a similar boiler having 880 square feet of heating surface would be rated at $\frac{880}{16} = 55$ horsepower.

Since the heating surface is only one of the factors entering into the quantity of steam generated per hour, it follows that a horsepower rating based on heating surface alone is of very small value as an aid in the selection of a boiler. About all that can be expected when buying a boiler according to this kind of rating is to receive an amount of heating surface depending on what ratio the maker of the boiler has adopted. The boiler, if thus bought, may or may not be suitable for the service it is to perform.

SIZE OF CHIMNEYS.

73. The formulas given below and the table calculated therefrom were first published by Mr. William Kent in 1884; they have since been extensively used and have met with much approval. The sizes given for the different commercial horsepowers are believed to be ample when the draft area through the boiler flues and connections is not less than 20 per cent. greater than the chimney area. When a number of boilers are connected to the same chimney, there

are usually long connections and a number of bends that increase the frictional resistances to the flow of the heated gases; in that case, both the height and the sectional area of the chimney may advantageously be increased.

Let A = cross-sectional area of the chimney in square feet;
H = height of chimney in feet;
X = commercial horsepower of boiler.

Rule 17.—*To find the commercial horsepower, subtract .6 times the square root of the area of the chimney from the area. Multiply the remainder by 3.33 times the square root of the height.*

Or, $$X = 3.33\,(A - .6\sqrt{A})\sqrt{H}.$$

Mr. Kent bases his estimate of the commercial horsepower of the boiler on a coal consumption of 5 pounds per indicated horsepower per hour; that is, he assumes that the plant will use 5 pounds of coal for each indicated horsepower developed by the engine or engines. Table IV has been calculated from rule **17**; hence, to find the number of pounds of coal burned per hour, multiply the figures in the table given under the heading "Commercial Horsepower" by 5.

Example.—Given, a chimney 100 feet high and having an area of 5 square feet; find the commercial horsepower of the boiler plant for which the chimney is adapted.

Solution.—Applying rule **17**, we get

$$X = 3.33\,(5 - .6\sqrt{5})\sqrt{100} = 122 \text{ H. P., nearly. Ans.}$$

74. A common rule for the area of the chimney is to make it $\frac{1}{8}$ to $\frac{1}{7}$ of the grate area. Then, the height of the chimney may be found by the following rule:

Rule 18.—*Multiply the horsepower by .3 and divide the product by the difference between the area and .6 times the square root of the area, and square the result.*

Or, $$H = \left(\frac{.3\,X}{A - .6\sqrt{A}}\right)^2.$$

TABLE IV.

SIZE OF CHIMNEYS AND HORSEPOWER OF BOILERS.

Height of Chimney in Feet.											Actual Area. Sq. Ft.	Side of Square. Inches.	Diameter. Inches.
50	60	70	80	90	100	110	125	150	175	200			
Commercial Horsepower.													
23	25	27									1.77	16	18
35	38	41									2.41	19	21
49	54	58	62								3.14	22	24
65	72	78	83								3.98	24	27
84	92	100	107	113							4.91	27	30
	115	125	133	141							5.94	30	33
	141	152	163	173	182						7.07	32	36
		183	196	208	219						8.30	35	39
		216	231	245	258	271					9.62	38	42
			311	330	348	365	389				12.57	43	48
			402	427	449	472	503	551			15.90	48	54
			505	539	565	593	632	692	748		19.64	54	60
				658	694	728	776	849	918	981	23.76	59	66
				792	835	876	934	1,023	1,105	1,181	28.27	64	72
					995	1,038	1,107	1,212	1,310	1,400	33.18	70	78
					1,163	1,214	1,294	1,418	1,531	1,637	38.48	75	84
					1,344	1,415	1,496	1,639	1,770	1,893	44.18	80	90
					1,537	1,616	1,720	1,876	2,027	2,167	50.27	86	96

Example.—For a 100-horsepower plant having 40 square feet of grate surface, using a ratio of grate surface to chimney area of 8 : 1, what height of chimney will be required?

Solution.—Chimney area $= \frac{40}{8} = 5$ square feet. Then, applying rule **18**, we get

$$H = \left(\frac{.3 \times 100}{5 - .6\sqrt{5}}\right)^2 = 67.24 \text{ ft.} \quad \text{Ans.}$$

It is recommended that the rules here given be not applied much beyond the ranges given in the table, since absurd proportions may then be obtained.

TABLE V.

CHIMNEY PROPORTIONS OF RETURN-TUBULAR BOILERS.

	Horsepower of Boiler.																	
	10	12	15	20	25	30	35	40	45	50	55	60	70	80	100	115	125	150
Diameter of boiler. Inches...	30	36	36	36	42	44	44	44	48	54	54	60	60	60	66	66	72	72
Length of boiler. Feet.....	8	7	8	10	10	10	12	14	14	12	15	12	14	16	16	18	16	18
Diameter of chimney. Inches...	14	16	16	16	20	22	22	22	24	26	26	28	28	28	30	30	34	34
Height of chimney. Feet....	24	24	28	35	35	35	40	50	50	40	50	40	50	60	60	60	60	60

75. The E. Keeler Company of Williamsport, Pennsylvania, for their return-tubular boilers, recommend the diameters and heights of chimneys given in Table V. In this table the rated horsepower of the boiler is one-fifteenth

of the heating surface. The size of chimney given is for a single boiler of the given dimensions, and the height is above the grate.

76. The same company recommends the heights and diameters of chimneys for single locomotive-type boilers given in Table VI, in which the rated horsepower of the boiler is one-tenth of the heating surface. In this table, the height of the stack is to be measured from the top of the boiler.

TABLE VI.

CHIMNEY PROPORTIONS FOR LOCOMOTIVE-TYPE BOILERS.

	Horsepower of Boiler.											
	15	20	25	30	40	50	60	70	80	100	125	150
Diameter of boiler. Inches	32	34	36	40	42	44	48	54	56	60	66	66
Length of firebox. Inches	36	42	48	48	48	54	60	60	60	60	60	66
Width of firebox. Inches	26	28	30	34	36	38	42	48	50	54	60	60
Length of tubes. Inches	84	90	96	96	120	132	138	144	144	168	180	192
Diameter of chimney. Inches	16	16	18	20	20	22	24	24	26	30	32	32
Height of chimney. Feet	24	24	24	24	30	36	40	40	40	46	50	55

77. The Phœnix Iron Works Company, Meadville, Pennsylvania, recommend the heights and diameters of chimneys for their horizontal return-tubular boilers given in Table VII. In this table the horsepower rating is based on an evaporation of $34\frac{1}{2}$ pounds of water from and at 212° F.

TABLE VII.

CHIMNEY PROPORTIONS FOR RETURN-TUBULAR BOILERS.

	Rated Horsepower.																		
	10	15	20	25	30	35	40	45	50	60	60	70	80	100	115	125	150	175	200
Diameter of boiler. Inches....	30	36	36	42	42	44	44	48	54	54	60	60	60	66	66	72	72	78	78
Length of boiler. Feet......	8	8	10	10	12	12	14	14	12	15	12	14	16	16	18	16	18	18	20
Diameter of chimney. Inches....	14	16	16	18	18	20	20	22	24	24	28	28	28	30	30	34	34	38	38
Height of chimney. Feet......	28	28	36	36	40	40	50	50	40	60	40	50	60	60	70	60	70	70	80

78. While the heights of chimneys given in Tables IV, V, VI, and VII are generally satisfactory for free-burning coal, they may have to be increased if finely divided fuel is to be burned. In general, the finer the fuel, the greater the intensity of the draft required, and, consequently, the higher the chimney must be. For anthracite down to pea coal, it is rarely advisable to use a chimney of less height than 80 feet; when smaller sizes or bituminous slack are to be burned, a height of 100 feet is not excessive. For culm, a chimney 150 feet would be needed. If these chimney heights cannot be installed, owing to the expense in first cost, mechanical draft will have to be resorted to.

79. Chimneys are usually built of brick, though in some cases iron stacks, which are often lined with brick, are

preferred. The external diameter of the base should be $\frac{1}{10}$ the height, in order to provide stability. The taper of a brick chimney is from $\frac{1}{16}$ to $\frac{1}{4}$ inch to the foot on each side. The thickness of brickwork is usually one brick (8 or 9 inches) for 25 feet from the top, increasing $\frac{1}{2}$ brick for each 25 feet from the top downwards. If the inside diameter is greater than 5 feet, the top length should be $1\frac{1}{2}$ bricks, and if under 3 feet, it may be $\frac{1}{2}$ brick, in thickness for the first 10 feet. The shell of iron stacks is generally the same size throughout its length; the lining, however, is made thinner from the bottom towards the top.

A round chimney is better than a square one, and a straight flue is better than a tapering one. If the flue is tapering, the area for calculation is measured at its smallest section. The flue through which the gases pass from the furnaces to the chimney should have an area equal to, or a little larger than, the area of the chimney. Abrupt turns in the flue or contractions of its area should be carefully avoided, as they greatly retard the flow of the gases. Where one chimney serves several boilers, the branch flue from each furnace to the main flue must be somewhat larger than its proportionate part of the area of the main flue.

SIZE OF SAFETY VALVE.

AREA OF SAFETY VALVE.

80. By the expression "area of safety valve," we mean the area of the opening in the valve seat, or, in other words, the projected area of the surface of the valve in contact with steam when the valve is closed.

The area of the valve should be at least large enough to discharge steam as fast as the boiler can generate it, for otherwise the steam pressure would rise even though the safety valve were open. Authorities differ greatly in their opinions regarding the area of valve that will be sufficient to fulfil this requirement.

81. Let G = grate surface in square feet;
P = steam pressure, gauge;
W = weight of steam generated per hour in pounds;
w = weight of coal burned per hour;
A = least area of safety valve in square inches.

The Bureau of Steam Engine and Steam Boiler Inspection of the city of Philadelphia and the Hartford Steam Boiler Inspection and Insurance Company prescribe the following formulas:

For natural draft,

$$A = \frac{22.5\,G}{P + 8.62}. \qquad (a)$$

For artificial draft,

$$A = \frac{1.406\,w}{P + 8.62}. \qquad (b)$$

The Board of Supervising Inspectors of Steam Vessels prescribe the following formulas:

For lever safety valves,

$$A = .5\,G. \qquad (c)$$

For pop valves,

$$A = \frac{G}{3}. \qquad (d)$$

Professor Thurston of Cornell University proposes

$$A = \frac{.5\,W}{P + 10}. \qquad (e)$$

The formulas here given are those in most common use, but will give widely varying results.

82. A comparison of the results obtained by using the different rules given is shown in the following example:

EXAMPLE.—With a boiler having a grate surface of 30 square feet, burning 400 pounds of coal per hour, and generating 3,000 pounds of steam per hour at 100 pounds gauge pressure, what should be the area of the lever safety valve?

SOLUTION.—Applying formula (*a*), Art. **81**, we get

$$A = \frac{22.5 \times 30}{100 + 8.62} = 6.21 \text{ sq. in. Ans.}$$

By formula (c) we have

$$A = .5 \times 30 = 15 \text{ sq. in.} \quad \text{Ans.}$$

By formula (e) we get

$$A = \frac{.5 \times 3,000}{100 + 10} = 13.64 \text{ sq. in.} \quad \text{Ans.}$$

Since different boiler inspectors and examiners of engineers usually have a preference for some particular rule, students when about to take an engineer's examination are advised to find out by inquiry what particular rule the examiner prefers. For an actual design, the rule used by the boiler inspector in the locality where the boiler is used must, of course, be taken in order to have the boiler pass the inspection.

For boilers not subject to legal inspection, formulas (a) and (b), Art. **81**, may be used.

83. Effective Area of Safety Valves.—By effective area is always meant the annular opening between the valve and its seat when the valve is lifted off its seat. This area depends on the angle of the seat and the vertical lift of the valve. It is the general practice to give the seat an inclination of 45° to the axis. The effective area may be calculated by the following rule,

where d = diameter of valve in inches;
l = lift of valve in inches;
e = effective area in square inches:

Rule 19.—*Add half the lift to the diameter of the valve. Multiply this sum by 2.221 and by the lift.*

Or, $$e = \left(d + \frac{l}{2}\right) 2.221\, l.$$

Rule **19** should be applied only to valves in which the seat makes an angle of 45° with the axis.

EXAMPLE.—What will be the effective area of a safety valve 4 inches in diameter with a vertical lift of .14 inch?

SOLUTION.—Applying rule **19**, we get

$$e = \left(4 + \frac{.14}{2}\right) \times 2.221 \times .14 = 1.2655 \text{ sq. in.} \quad \text{Ans.}$$

EXAMPLES FOR PRACTICE.

1. What area of safety valve would be required for a boiler burning 380 pounds of coal per hour under forced draft and carrying 80 pounds per square inch steam pressure? The boiler is located in Philadelphia, Pennsylvania. Ans. 6.03 sq. in.

2. What is the effective area of a 5-inch safety valve when lifted .14 inch off its seat? Ans. 1.5765 sq. in.

REENFORCING RINGS.

INTRODUCTION.

84. Purpose.—When manholes are cut into the shell of cylindrical boilers, the opening must be reenforced either by one or by two reenforcing rings in order to make up for the loss of strength occasioned by the cutting of a large hole through the sheet. When only one reenforcing ring is used, it may be placed either inside or outside of the shell; when two rings are used, one is placed inside and one outside of the shell. The thickness of a single reenforcing ring may be from 1.25 to 1.5 times the thickness of the shell plate; when two rings are used, the thickness of each may be equal to the thickness of the shell.

Reenforcing rings must be securely riveted to the shell, and the rivets must be so proportioned and their number must be such that their combined resistance to shearing will be at least equal to the resistance of the reenforcing rings to a tensile stress. The rings should be made of the same material as the shell.

RULES FOR DESIGNING REENFORCING RINGS.

85. Symbols.—

Let w = width of reenforcing ring;
t = thickness of reenforcing ring;
d = diameter of rivet when driven;
t_1 = thickness of shell plate;

T = tensile strength of the ring per square inch of section;
a = net section of the ring;
S = shearing strength of rivet per square inch of section;
l = length of opening in shell;
n = number of rivets.

86. Rule 20.—*To find the width of the reenforcing ring when but one ring is used, multiply the length of the opening in the shell in the direction of the length of the boiler by the thickness of the shell; divide this product by twice the thickness of the reenforcing ring and add the driven diameter of the rivet to the quotient, if one row of rivets is to be used. If the ring is to be double-riveted, add twice the driven diameter of the rivet to the quotient.*

Or, $$w = \frac{l\,t_1}{2\,t} + d \text{ for a single-riveted ring,}$$

and $$w = \frac{l\,t_1}{2\,t} + 2\,d \text{ for a double-riveted ring.}$$

Example.—A manhole opening is 11×15 inches, measuring 11 inches in the direction of the length of the boiler. If the shell plate is $\frac{1}{2}$ inch thick and a single-riveted reenforcing ring $\frac{5}{8}$ inch thick is to be used, how wide should it be? The rivets are to be 1 inch driven size.

Solution.—Applying the rule just given, we get

$$w = \frac{11 \times \frac{1}{2}}{2 \times \frac{5}{8}} + 1 = 5.4 \text{ in. Ans.}$$

87. Rule 21.—*To find the width of a reenforcing ring when two rings are used, multiply the length of opening in the shell in the direction of the length of the boiler by the thickness of the shell; divide this product by 4 times the thickness of the rings and add the driven diameter to the quotient for single-riveting. For double-riveting, add twice the driven diameter of the rivet.*

Or, $$w = \frac{l\,t_1}{4\,t} + d \text{ for single-riveted rings,}$$

and $$w = \frac{l\,t_1}{4\,t} + 2\,d \text{ for double-riveted rings.}$$

EXAMPLE.—A manhole opening 12×16 inches, measuring **16** inches in the direction of the length of the boiler, is to be reenforced with two rings $\frac{3}{4}$ inch thick. The shell plate being $\frac{3}{4}$ inch thick and 1 inch rivets (nominal size) to be used, what should the width of the rings be for single-riveting?

SOLUTION.—The driven size of rivet $= 1 + \frac{1}{16} = 1\frac{1}{16}$ inches. Applying rule **21,** we get

$$w = \frac{16 \times \frac{3}{4}}{4 \times \frac{3}{4}} + 1\tfrac{1}{16} = 5\tfrac{1}{16} \text{ in. Ans.}$$

88. In the calculation of the minimum number of rivets, the net section of the ring is used. For single-riveting, the net section is found by subtracting the driven diameter of the rivet from the width of the ring and multiplying the remainder by the thickness of the ring. For double-riveting, subtract twice the driven diameter of the rivet from the width of the ring and multiply the remainder by the thickness of the ring.

The trial nominal diameter of the rivets for a single reenforcing ring may be made about equal to the thickness of the shell plate $+ \frac{7}{16}$ inch; for two reenforcing rings it may be made about equal to the thickness of the shell plate $+ \frac{5}{16}$ inch.

89. Rule 22.—*To find the number of rivets for a single reenforcing ring, multiply the net section of the ring by 4 times the tensile strength of the material and divide this product by the product of the shearing strength of the rivet and its area.*

Or,
$$n = \frac{4\,T\,a}{S\,d^2 \times .7854}.$$

EXAMPLE.—How many rivets $\frac{13}{16}$ inch in nominal diameter are to be used for a single reenforcing ring $\frac{1}{2}$ inch thick and 4 inches wide? Take the tensile strength of the ring as 60,000 pounds and the shearing strength of the rivets as 38,000 pounds per square inch of section. The reenforcing ring is to be single-riveted.

SOLUTION.—The driven size of the rivet is $\frac{13}{16} + \frac{1}{16} = \frac{7}{8}$ inch. The net section of the ring is $(4 - \frac{7}{8}) \times \frac{1}{2} = 1.56$ square inches. Applying rule **22,** we get

$$n = \frac{1.56 \times 4 \times 60{,}000}{38{,}000 \times (\frac{7}{8})^2 \times .7854} = 17. \quad \text{Ans.}$$

90. Rule 23.—*To find the total number of rivets for a pair of equal reenforcing rings, multiply the net section of one ring by 8 times the tensile strength of the material, and divide this product by the product obtained by multiplying together 1.85 times the shearing strength of the rivet per square inch of section and the area of the rivet.*

Or, $$n = \frac{8\,T\,a}{1.85\,S\,d^2 \times .7854}.$$

EXAMPLE.—In a manhole reenforced by a pair of reenforcing rings, the rings are $\frac{3}{4}$ inch thick and $4\frac{1}{4}$ inches wide. With single-riveting, how many $\frac{15}{16}$-inch rivets should be used? Take the tensile strength of the rings as 60,000 pounds per square inch and the shearing strength of the rivets as 38,000 pounds per square inch.

SOLUTION.—The driven size of the rivet is $\frac{15}{16} + \frac{1}{16} = 1$ inch. The net section of the ring is $(4\frac{1}{4} - 1) \times \frac{3}{4} = 2.44$ square inches, nearly. Applying rule **23**, we get

$$n = \frac{8 \times 60,000 \times 2.44}{1.85 \times 38,000 \times 1^2 \times .7854} = 21 \text{ rivets. Ans.}$$

91. When it is believed that the number of rivets given by an application of rules **22** and **23** is too small, so that the rivets will be too widely spaced, a smaller diameter of rivet may be chosen and the proper rule applied again to find the number of rivets. Conversely, when the number of rivets becomes too large, so that they will come too close together, a larger diameter of rivet should be chosen and the proper rule applied again.

Reenforcing rings whose width is made as called for by rules **20** and **21** will have a total net cross-sectional area equal to the cross-sectional area of the metal removed from the shell. By total cross-sectional area is here meant twice the net sectional area of the ring for a single ring, and 4 times the net sectional area of one ring when two rings are used. The rules for the number of rivets make the resistance of the rivets to shearing equal to the resistance to a tensile stress of the ring or rings. There is nothing to be gained by making the rivets more numerous than called for by rules **22** and **23**.

ECONOMIC COMBUSTION OF COAL.

GENERAL PRINCIPLES OF COMBUSTION.

1. Since fuel, chiefly in the form of coal, is the source of all the energy made available by the steam engine, a study of the principles governing the economical generation of power by means of the steam engine must begin with a study of the combustion of fuel and the means of preventing a waste of heat by guarding against those conditions that tend towards incomplete combustion. Heat lost by imperfect combustion cannot be recovered by any means; no arrangement of elaborate and economical machinery in the engine room will utilize the energy wasted in the fireroom through ignorance or carelessness on the part of the fireman. Furthermore, one of the most serious problems confronting the steam engineer is the burning of soft coal without the formation of black smoke, and this problem depends for its solution more on intelligent work in the fireroom, which involves a knowledge of economic combustion, than on patented furnaces.

2. A general outline of the properties of the more important fuels and the elementary principles of combustion have already been given, but in order to better understand the principles underlying complete and economical combustion and smoke prevention, a more detailed study of the physical and chemical changes occurring in the furnace will now be made.

PHYSICAL AND CHEMICAL OCCURRENCES OF COMBUSTION.

3. Bituminous coal is composed largely of various compounds of carbon and hydrogen called **hydrocarbons.** When the coal is heated, these compounds are driven off partly in the form of permanent gases and partly as vapors that may easily be condensed or changed to a liquid form. The process of separating the gases and vapors from the part of the coal that cannot be vaporized by the mere action of heat is called **distillation;** the substances driven off from the coal by heat are called **volatile substances,** while the portion remaining forms **coke,** which is composed chiefly of carbon. This carbon is called the **fixed carbon** of the coal.

4. Classification of Volatile Substances.—The volatile substances may be divided into two classes, viz., **non-combustible** and **combustible** substances.

The first class consists mostly of water, free oxygen, and nitrogen; these are driven off when the coal is heated—the water as steam and the gases in their free state.

The second class consists of the hydrocarbons, which comprise numerous compounds of hydrogen and carbon. When the coal is heated, part of the hydrocarbons are driven off in a gaseous form and part as vapors.

5. The principal gases in the volatile combustible are **carbureted hydrogen,** or **marsh gas,** consisting of 1 atom of carbon and 4 atoms of hydrogen, as shown by its symbol CH_4, and **olefiant gas,** C_2H_4. With many coals, free hydrogen is given off in considerable quantities; small quantities of other less important gases are also generally present.

The vapors are mostly coal tar and naphtha, with small quantities of sulphur. Their presence can be detected by placing a cold iron bar into the yellow gases rising from a fresh charge; a sticky coating, consisting mostly of the condensed tar, will form on the cold metal.

6. The proportion of the volatile matter in coal depends on its composition. Anthracite consists almost entirely of fixed carbon and ash; in some bituminous coals, the greater part is volatile. The relative proportions of fixed gases and condensible vapors in the volatile parts of coal also vary with the composition of coal. In some cases the volatile matter contains considerable quantities of the tarry vapors and heavy hydrocarbon gases, as C_2H_4, while in others it consists largely of the light marsh gas CH_4 and free hydrogen.

The quantity and composition of the volatile matter depends not only on the composition of the coal itself, but also on the conditions under which distillation takes place, a difference in the temperature and the presence or absence of air or steam modifying the composition of the vapor and gases to a very great extent. Irregular firing and draft result in a great difference in the quantity and composition of the gas burned in the furnace at different periods.

7. With few exceptions, coal contains small quantities of sulphur, usually in combination with some other element; one of the most common compounds is that of sulphur and iron, known as **iron pyrites.** When the coal is heated, the sulphur is separated from the iron and burns to sulphur dioxide, SO_2. The heat derived from the combustion of the sulphur found in coal is small, but the sulphur dioxide formed, in combination with the moisture in the gases, corrodes iron very rapidly; any relatively cold metal exposed to the gases from coal rich in sulphur is rapidly corroded and destroyed.

8. Conditions Involved in the Combustion of Volatile Substances.—At the temperature generally existing in a boiler furnace, the tar and other liquids vaporize and mix with the gases. This gaseous mixture is readily burned under proper conditions of air supply and temperature. The conditions involved in the combustion of the gases and vapors may readily be studied by the aid of the flame of a common tallow candle. The tallow forms a supply of

solid hydrocarbon that, under the action of the heat of the flame, is liquefied; then, by capillary attraction, it is drawn up into the wick to a point where the heat is sufficient to vaporize it. A process of distillation here goes on, and a blue, transparent, cone-shaped mass of vapor and gas, shown at c, in Fig. 1, is formed. The heat of the flame causes a current of air to approach from all sides and provides a supply of oxygen that comes in contact with the surface of the hot cone of vapor. The air that reaches this part of the flame, however, is not sufficient for the complete combustion of the vapor; and since the hydrogen in the hot cone has a greater affinity for oxygen than it has for carbon, it combines with all the available oxygen and burns to water, H_2O. The carbon is left in the form of minute solid particles that become highly heated and give this part of the flame its bright yellow color. As the hot particles of carbon rise, they come in contact with the inward current of air, which furnishes oxygen sufficient to burn them to CO_2. The gas so produced, mixed with the H_2O from the luminous portion b of the flame, forms the nearly colorless outer and upper section a that gradually mixes with the surrounding air, cools, and becomes wholly invisible.

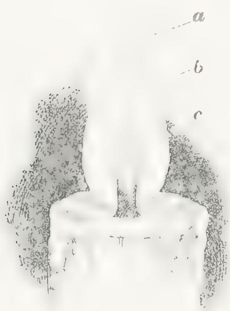


FIG. 1.

9. If we insert the end of a glass tube into the inner cone of the flame, as shown in Fig. 2, a part of the gas that has not yet received a supply of air may be drawn off. In passing through the tube, this gas is cooled below the temperature at which it will burn and issues from the tube into the air without igniting. By applying a lighted match to this gas and heating it, it may be lighted and burned. This simple experiment illustrates two of the most important principles involved in the economical combustion of the gases in a boiler furnace. It shows: *First*, that the gas

that has been cooled below the temperature of the ignition cannot be burned merely by furnishing a supply of air. *Second*, that when the gas is supplied with air, it can be burned if its temperature is raised to the igniting point.

FIG. 2.

10. Igniting Temperature of Carbon and Volatile Substances.—In order to burn the fixed carbon that is in the coal, a high temperature is needed to cause the atoms of carbon to combine with the oxygen supplied by the air. The igniting temperature of the fixed carbon and also of the volatile substances is estimated to be about 1,800° F. Since the maximum temperature in the furnace rarely exceeds 2,500° F. and is ordinarily several hundred degrees less, it is seen that on account of the relatively small difference between the igniting temperatures of the carbon and gases and the maximum temperature in the furnace, constant care is needed to prevent the temperature in the furnace falling below 1,800° F. The temperature in the furnace can be judged quite accurately by the appearance or color of the fire in accordance with the relation between color and temperature given in the accompanying table.

When the supply of air is sufficient and the temperature high enough, the carbon burns to carbon dioxide, CO_2, which, being the product of complete combustion, is incombustible. With a high temperature and deficient air supply, carbon monoxide, CO, is formed. Since this gas is the product of incomplete combustion, it can be burned to CO_2 by bringing it into intimate contact with air while highly heated.

RELATION BETWEEN COLOR AND TEMPERATURE.

Temperature. Degrees Fahrenheit.	Appearance.	Temperature. Degrees Fahrenheit.	Appearance.
980	Red—just visible	2,010	Dull orange
1,290	Dull red	2,190	Bright orange
1,470	Dull cherry red	2,370	White heat
1,657	Full cherry red	2,550	White, welding heat
1,830	Bright red	2,740	White, dazzling heat

11. Chemical Occurrences of Burning Solid Carbon. When solid carbon burns on a grate, the chemical changes or reactions are about as follows: Some of the oxygen of the air that rises through the grate combines with the first layers of hot carbon in the proportion of 2 atoms of oxygen to 1 atom of carbon, and carbon dioxide, CO_2, is formed. As the gases rise through the fire, more of the oxygen combines with carbon, and as long as the supply of air is sufficient and well distributed, the combination is mostly in the proportion that produces CO_2. With a thick bed of fuel, however, or an arrangement of the fuel that does not permit of a proper distribution of the air, there will be some portions of the fire in which the supply of oxygen is not great enough to furnish the 2 atoms for the production of CO_2; only 1 atom of oxygen will be available for combination with some of the carbon atoms burned, and the product will be CO. Further, when a molecule of CO_2 comes into close contact with the hot carbon, the attraction of the carbon for oxygen is so great that 1 atom of the oxygen leaves the CO_2 and combines with an atom of carbon; 2 molecules of CO are thus formed, one by the separation of one of the oxygen atoms from the molecules of CO_2 and the other by the combination with an atom of

carbon of the oxygen atom so released. The separation of a given weight of CO_2 into CO and O absorbs as much heat as was developed when the CO combined with oxygen to form the CO_2; the net production of heat is the same whether a certain amount of carbon is burned to CO directly and passes off in that form, or a part of it is first burned to CO_2 and this gas is then decomposed with the production of CO and oxygen that combines with the remainder of the carbon to form CO. The carbon monoxide formed in the fuel bed passes into the furnace, and if there is not sufficient oxygen present or if the temperature is not high enough, it will pass away unburned. If sufficient oxygen is present and the furnace temperature is high enough, each molecule of CO will combine with another atom of oxygen and thus burn to CO_2.

12. A careful study of the above outlines of the processes involved shows that economical combustion, both of the volatile matter and of the solid carbon, involves the following essential conditions: (1) There must be a supply of air sufficient to furnish the oxygen required for complete combustion. (2) This air must be so distributed as to bring the oxygen into intimate contact with all parts of the fuel. (3) The temperature must be high enough to bring about the combustion. *With either of these essentials lacking, there will be incomplete combustion and a loss of heat.*

SMOKE.

CLASSIFICATION OF SMOKE.

13. The smoke arising from an ordinary fire may properly be divided into two classes: *First*, the partly condensed tarry vapors produced by the distillation of the fuel; *second*, the minute particles of solid carbon left when the hydrocarbons are but partially burned. Smoke of the second class is so objectionable that in many cities there

are special ordinances that either prohibit the burning of soft coal or require it to be burned in such manner that there is no formation of smoke.

FORMATION OF SMOKE.

14. When hydrocarbons are heated in the presence of air, the affinity of the hydrogen for oxygen is great enough to cause it to separate from the carbon and combine with the oxygen. If the supply of air is sufficient and the temperature is great enough, the carbon set free will burn and no smoke will be produced. If, however, there is a limited supply of air or too low a temperature for their combination with oxygen, the carbon atoms will combine with each other and form molecules of carbon that collect into minute particles of solid carbon. It is the presence of these hot-carbon particles that gives color to the flame of an ordinary lamp or fire. If they can be supplied with oxygen at a high enough temperature, they will burn, as we saw in the flame of the candle, Fig. 1. Under unfavorable conditions, however, these solid particles of carbon do not burn, but are deposited as soot or are carried out of the furnace with the gases as smoke. The nature of these unfavorable conditions can well be studied by considering the action of an ordinary kerosene lamp. With a good chimney, a flame not too high, and a clean burner, the lamp burns with a bright, steady flame and no smoke. The bright flame indicates a high temperature and the absence of smoke shows that the air supply is ample and well distributed. Insert a cold iron rod into the flame from the top of the chimney; a deposit of soot—unburned carbon—collects on the rod and the gas that rises along the sides of the rod is cooled so much that a part of the carbon is not burned and considerable smoke is formed. Interfere with the air supply by partly closing the top of the chimney or the holes in the burner; a cloud of smoke is formed. Remove the chimney, so as to interfere with the distribution of the air to the lower part of the flame while a large volume of cold air meets the upper part, and we have

a case in which the carbon is cooled by the action of an excessive supply of cold air imperfectly distributed. Turn the wick too high, and there is more gas formed than can be supplied with air before it gets too cold for the carbon to burn.

SMOKE PREVENTION.

15. It is seldom that the supply of air to a boiler furnace is not great enough to burn the carbon and prevent the formation of smoke; in fact, it is often found that large volumes of smoke are accompanied by a liberal supply of oxygen. The more common condition when smoke is formed is too low a temperature of the fire or a distribution of the air that prevents oxygen from reaching the carbon before it is cooled by contact with the boiler plates. Heavy firing, by means of which large volumes of gas are formed and the furnace greatly cooled, is one of the most prolific causes of smoke production. Owing to the high ignition temperature of solid carbon, smoke, when once formed, is extremely difficult to burn. With properly constructed furnaces, good draft, and careful management *smoke prevention* is possible; *smoke consumption*, however, may be said to be practically impossible under any of the conditions existing in the furnace of a steam boiler.

16. If the bed of burning fuel is thick and the draft sluggish, carbon monoxide will rise from the surface of the fire, and if it is not burned to carbon dioxide by mingling with air admitted above the fire, it will pass through the flues and cause a great loss of heat, about 10,100 B. T. U. for each pound of carbon.

Two remedies for this type of loss are available. *First*, the fuel bed may be made so thin that sufficient air can enter through the grate either to prevent the formation of carbon monoxide or to burn it to carbon dioxide as soon as it is formed. *Second*, sufficient air to burn the carbon monoxide that rises from the coal may be admitted into the furnace above the fire. If there is a brisk fire and a high temperature in the furnace, the carbon monoxide can be easily

burned by either of the two methods given, a short, pale blue flame being produced near the surface of the fuel. With a slow, dull fire, the temperature in the furnace may be too low to ignite the escaping gas, which will pass out with any air that may be present, no combination taking place. There is, thus, a double loss; heat is wasted not only by the escape of carbon monoxide, but also by the heating of the excess of air.

17. The presence of unconsumed carbon monoxide is not easily detected by mere observation, owing to the gas being colorless. By regulating the fire, however, so that a high temperature is maintained in the furnace, and by watching the surface of the burning fuel to see that sufficient air is present to produce the pale blue flame previously mentioned, we can make reasonably sure that the amount of unconsumed carbon monoxide will be exceedingly small.

With a large grate and a very low rate of combustion, there is usually considerable formation of unconsumed CO, owing to the relatively low temperature in the furnace. The remedy is to restrict the grate area by bricking up with one or more courses of firebrick so as to secure a higher rate of combustion and the consequent high temperature. Simply admitting more air, when there is a very low rate of combustion, is liable to aggravate the heat losses, since it will result in still further lowering the temperature of the furnace.

The escape of unconsumed hydrocarbon, with its attendant loss of heat, is more readily detected by direct observation of the fire than that of carbon monoxide, especially when the loss from this source is large. The yellowish vapors that rise from freshly fired coal are familiar to all firemen and are an evidence of unconsumed hydrocarbons. When the heavier hydrocarbons are contained in the smoke in considerable quantities, the smoke from the chimney is yellowish in color and has a tarry odor.

18. Conditions for Complete Combustion of Hydrocarbons.—The conditions required for the complete

combustion of the hydrocarbons are the same as for the fixed carbon of the fuel; that is, a sufficient air supply, an intimate mixture of the air with the gases, and a high temperature. In practice it seldom happens that a large part of the carbon in the volatile matter escapes unburned, even when a great deal of black smoke is produced. Owing to its finely divided state, a small quantity of carbon will give a high color to a large volume of gas coming from a chimney, and there will appear to be a serious waste when the actual heat loss is really comparatively small.

Numerous careful tests have shown that the production of black smoke does not necessarily represent a great loss of efficiency. In many cases it has even been found that a reduction of efficiency accompanied the use of furnace arrangements that were successful in preventing smoke. The reason for this appears to be that the gases were burned with a large excess of air, which while so controlled as to burn the carbon to CO or CO_2, thus rendering it invisible, carried much more heat from the furnace than was developed by the comparatively small weight of carbon that would have appeared in the smoke. The losses due to solid carbon falling into the ash-pit, the escape of unburned hydrocarbons, incomplete combustion of the fixed carbon on the grate, and a needless excess of air are always much more serious than the loss due to the formation of black smoke.

Although smoke in itself represents no serious loss of heat, its production in large quantities is often the result of conditions that produce imperfect combustion of the carbon monoxide and the hydrocarbons. A low furnace temperature, the piling of coal on the fire in such large quantities that an excessive volume of combustible gas is given off while, at the same time, the furnace temperature is lowered by the heat absorbed in the process of distilling the volatile matter from the freshly fired coal, and also the admission of large volumes of cold air tend to produce black smoke and, besides, represent serious and unnecessary heat losses.

19. Effect of Moisture on Combustion.—There are processes accompanying the combustion of coal in the boiler furnace that in themselves absorb great quantities of heat and, in consequence, have an important bearing on the question of complete and economical combustion. All moisture that enters the furnace with the fuel must be evaporated at the expense of the heat developed by the combustion of the coal; the vapor thus formed passes out of the chimney at a temperature seldom less than 400 degrees. Assuming that the moisture enters the furnace at a temperature of 70° and leaves at 400°, 1,200 B. T. U., nearly, will be lost for each pound of moisture in the coal. Moisture in coal can only be driven off by heating it above ordinary temperatures, but it will be readily absorbed at ordinary temperatures; hence, it is important that coal when stored is not exposed to rain—it should always be stored under cover. In some cases it may be advantageous to wet bituminous coal; when wet, especially if the coal is fine, it cokes better and there is, thus, less waste from coal falling into the ash-pit. Wetting is only recommended for bituminous slack and anthracite culm, and should be as moderate as will secure the results desired. It would be much better, however, when the draft is moderate to use grates having smaller openings. When the draft is exceptionally strong and very fine coal is burned, wetting becomes almost a necessity. If not done, the strong draft will actually carry a large percentage of the fine coal up the stack.

20. The distillation of the volatile matter is a process that, with all bituminous coals, and to a lesser degree with anthracite, absorbs a great deal of heat. The effect of this absorption of the heat is clearly shown by the drop in the steam pressure when a large quantity of fresh coal is thrown on the fire, which drop is largely due to the lowering of the furnace temperature through the heat absorbed by the distillation of the volatile matter. While the absorption of heat is necessary to drive off the volatile combustible, the losses attendant to a lowering of the furnace temperature can be minimized by frequent light charges of coal.

METHODS OF SECURING AN AIR SUPPLY.

21. It has been shown that the complete and economical combustion of coal and the prevention of smoke depend primarily on a sufficient air supply being brought into intimate contact with the fixed carbon and volatile combustibles. Without the proper distribution of the air supply, the high temperature necessary for complete combustion cannot be maintained; therefore, different methods of securing this distribution will now be considered.

22. Air Supply Below the Grate.—One of the best methods of supplying the air is that in which the fire is thin and open enough to permit sufficient air to rise through the grate and fuel bed. The advantages of this method are as follows:

The air, in rising through the fuel bed, becomes highly heated. If a clean, even fire is maintained, the air supply is well distributed and comes in intimate contact with the gases almost as soon as they are formed; the air and gases are thus thoroughly mixed in the vicinity of the hottest part of the furnace and complete combustion follows. The danger of chilling the boiler plates by a current of cold air is less than when the air is admitted at any point above the grates.

23. With this method of air supply, the firing must be carefully done; the fire must be maintained at a moderate and even thickness, the grates must be kept clean, and each air space must be kept as free from clinkers as possible. No bare spots should be allowed to form, and, finally, the coal should be supplied in small quantities and often, each shovelful being spread over as much surface as possible. Large lumps should always be broken before being fired, and in no case should a thick mass of freshly fired coal be allowed to collect in any part of the furnace. Such a method of firing demands care and close attention on the part of the fireman, but if it is carefully followed, the coal will be burned in a very economical manner and without the formation of black smoke. The reduction in the amount of coal that must

be handled will, in a great degree, make up for the apparent increase in labor connected with such a system of firing.

The thickness of the bed of fuel to be used with this system of firing depends on the quality of the fuel and the intensity of the draft. In general, it may be stated that the best results will be obtained with a good strong draft that will permit of a fuel bed, with reasonably good bituminous coal, of from 7 to 12 inches in thickness. The thickness can best be determined by actual trials and carefully watching the results.

24. Air Supply Above the Grate. In a great many cases it is impracticable, or at least difficult, to regulate the fire so carefully as to secure a proper supply of air through the grates. For example, in locomotive work, where the demand for steam and the intensity of the draft are very irregular, it is found to be practically impossible to maintain a depth of fire that will conform to the irregularity in the conditions of running. The heavy currents of air drawn through the thin bed of fuel by the irregular action of the exhaust tear holes in the fire if it is kept too thin, thus allowing large quantities of cold air to enter at one place; these currents of cold air, in addition to their evil effects on the combustion of the gases and their chilling effect on the boiler, often carry considerable quantities of solid fuel into the flues.

It is seldom that the conditions are as unfavorable in stationary practice as in locomotive work, but there are many cases in which the irregularity of the draft or of the demand for steam are considerable; the quality of the fuel may also make it very difficult to keep a clean, open fire that will permit the steady supply of air demanded; it is therefore often essential that means be provided for furnishing at least a part of the air required to burn the gases through openings above the grate. Of the methods used for this purpose, the one most commonly found is, perhaps, either a partly opened fire door or some special arrangement of openings through the fire door that serve as an inlet for the air and distribute it over the fire in a more or less satisfactory manner.

A partly opened door must be regarded as one of the most unsatisfactory means of getting air into the furnace that could be devised. The air enters in a large stream that is more likely to cool the small proportion of the gas that it reaches below its ignition temperature than it is to promote its combustion; further, the concentrated current of cold air is almost sure to strike a limited section of the furnace, which section becomes chilled; this, in turn, produces severe stresses in the boiler plates. Much more satisfactory results are obtained when the door is provided with special openings that serve to divide the entering current of air and distribute it over a considerable part of the grate. The perforated inner door is particularly useful in this respect; it serves the combined purpose of protecting the outer door from the heat radiated from the fire and of dividing the entering current of air into a large number of small jets that are considerably heated in their passage over its surface and are then distributed to the fire through the perforations and around the edges of the plate.

25. With a large furnace, it may be difficult either to admit sufficient air through the door to supply all parts of the furnace or to properly distribute the air so admitted. To remedy this defect, various methods of introducing the supply at the sides of the furnace and at the bridge wall have been devised. In locomotive boilers, a few rows of hollow staybolts are sometimes used just above the surface of the fire; these admit air in small jets that enter that part of the furnace least likely to be reached by air from the door, at right angles to the current of the gases, and the air is thus more easily mixed with the gas than would be the case if the currents were parallel.

26. Supplying Air to Externally Fired Boilers.—A method employed in furnaces for externally fired boilers is the use of hollow tiles that lead air to the side of the firebox and deliver it in small jets through openings in the brickwork above the surface of the fire. In many cases the conduits through which the air enters are placed in positions

that are intended to insure the heating of the entering air, thus delivering it at a high temperature and promoting combustion. So far none of these special devices for supplying air to the sides of the furnace have been sufficiently successful to secure their extensive use. It seems that their advantages are much more apparent than real. There can be no great gain in any attempt to heat the air, since the quantity that must flow through a given passage will be so great that its temperature will be but little elevated, unless the conduit is longer or the temperature to which it is subjected is greater than is practicable in the construction of most furnaces.

Another point that would seem to be unfavorable to the success of any attempt to introduce air through the sides of the furnace is the difficulty of controlling the quantity so introduced in accordance with the varying conditions of draft and fuel supply.

27. Air Supply in Bridge Wall.—Numerous methods of introducing air through openings in the bridge wall have been devised, some of which have been put into more or less successful practice. In many of these devices attempts have been made to heat the air so introduced by passing it through some part of the furnace or setting in which there is a high temperature. Such attempts, however, have probably never resulted in sufficient benefit to pay for the expense involved in the more complex construction of the furnace.

The admission of air through properly constructed openings in the bridge wall and above the fire has been shown, by experiment, to produce economical results when a bituminous coal, rich in volatile matter and producing a long smoky flame, is used. The openings through the bridge wall should be so arranged as to discharge the air in a number of jets at a right angle to the direction of flow of the gases. The space back of the bridge should then be large enough to form a combustion chamber in which there will be a thorough mixture and complete combustion before the air and gas are chilled by entering the tubes. A large space

produces a relatively moderate velocity of flow of the gases, which is favorable to the intimate mixture of the air and gas.

28. Introduction of Air at High Temperatures.—While it would be an advantage to have the air enter at a high temperature, no means have yet been devised that will accomplish this in an economical manner. Of course, the temperature of the air is raised to a slight degree in its passage through the ash-pit, or the openings in the setting leading to the bridge, and in passing through the bridge to the openings through which it is discharged, but the actual gain by this means is relatively unimportant, even when rather elaborate systems of passages are used.

29. Some of the difficulties attendant on this method are the following:

In order to prevent loss from too much or too little air, the passages must be controlled by dampers that should be regulated according to the condition of the fire and the rate of combustion; this demands close attention and intelligent care on the part of the fireman, who must either maintain nearly constant conditions in the furnace or change the dampers as often as there is a material change in the condition of the fire. The passages through the bridge wall may give trouble by becoming clogged with ashes and clinkers pushed back from the fire during cleaning or carried back by the draft, and so become useless. The method is limited in its application to furnaces so constructed that there is room for a combustion chamber of considerable size, in which mixture and combustion may take place before the gases are cooled.

SHAPE AND SIZE OF FURNACE.

30. The lowest temperature at which ignition of the gases can take place is about 1,800°; by reference to the Steam Table, it is seen that the temperature of the water in the boiler, even under a pressure of 200 pounds per square inch, is less than 400°; this is practically the temperature of

the surface of the boiler. It is therefore evident that any gas that comes into close contact with the boiler before being burned will be cooled below the point of ignition, and, unless subsequently heated, will be carried to the chimney unconsumed. With coals containing large quantities of volatile matter and burning with a long smoky flame, it follows that the firebox must be of ample depth to provide a space in which the great volume of gas can burn before being cooled; or there must be a combustion chamber in which the gas can burn after leaving the firebox.

The danger of loss from cooling the gases too suddenly is greatest in internally fired boilers of the vertical and locomotive type. Unless the crown sheet is unusually high above the grate, vertical boilers are especially unfitted for the use of rich bituminous coals. Locomotive boilers, if properly managed, are better; by the use of the coking system of firing, the gas must pass through the length of the firebox and over the hot bed of coke, thus giving it considerable time to burn before entering the tubes, the heat radiated from the coke helping to keep it at a temperature at which combustion is possible.

31. Firebrick Arches.—Many locomotive boilers are fitted with **firebrick arches** that extend from the tube sheet towards the door and force the gases to take a path, first towards the door and then back above the arch to the tubes. The arch becomes highly heated, thus preventing the cooling of the gases before they become mixed with the air, and the path of the gases is lengthened sufficiently to enable them to burn before entering the tubes. It also increases the life of the flues by preventing the entrance of large volumes of cold air when the door is opened, and thus aids in maintaining a more even temperature.

The opinion of railroad men, almost without exception, is in favor of the firebrick arch for locomotives burning bituminous coals, and there is no doubt that the principle involved in its use can be employed to great advantage in stationary practice. Examples of modifications of the

principle of the firebrick arch are seen in some of the automatic stokers now in use and in the furnaces of such boilers as the Stirling water-tube boiler. In these furnaces, however, the position of the arch over that part of the grate nearest the fire door, while it helps to keep up a high and equable furnace temperature, does not materially aid in securing a more satisfactory mixture of gas and air, nor does it lengthen the path of the gas generated in that part of the furnace nearest the boiler. For these reasons, furnaces of this class are best adapted to the coking system of firing.

THE COMBUSTION CHAMBER.

32. In internally fired boilers of the Cornish, Lancashire, Scotch Marine, and similar types having large furnace flues that open into a chamber in the rear, the gases that are cooled by contact with the walls of the flue and pass through it unconsumed are sometimes burned in the rear chamber, which is given the name **combustion chamber** for this reason. In order to be effective, the combustion chamber should be provided with a mass of firebrick that can be maintained at a sufficiently high temperature to heat the gases to the ignition point. The currents of gases from the flues must then be brought into contact with this hot brickwork and thus burned before entering the tubes. These are conditions that are maintained with difficulty, especially in boilers, like the Cornish, Lancashire, and Galloway, having long flues in which the gases are considerably cooled before reaching the combustion chamber.

With boilers of the return-tubular type, the space back of the bridge serves the purpose of a combustion chamber to a certain extent. The gases, however, tend to rise and flow along the cold surface of the boiler; it is, therefore, difficult to prevent a considerable body from thus becoming cooled below the ignition temperature and passing unconsumed into the tubes. To overcome this difficulty, various arrangements of brick arches and flues have been placed between

the bridge and the back end of the boiler, the purpose of these devices being to present heated surfaces of brick to the current of the gas and to promote the intimate mixture of gas and air. Some of them are moderately effective, but they are subject to the limitations that were referred to in the case of the combustion chambers of internally fired boilers; that is, there is danger that the temperature of the brickwork will not be high enough to secure the ignition of any great proportion of the gases that pass beyond the bridge, or that much of the gas will not be brought into close enough contact with the hot bricks to be heated to the ignition point.

33. The construction of the type of furnace commonly used with water-tube boilers of the Babcock and Wilcox type is such that there is little opportunity for combustion to take place after the gases leave the firebox. The gases rise nearly vertically from the fuel bed and pass from the firebox immediately into contact with the tubes; the narrow spaces between the tubes divide the gases into thin sheets that are rapidly cooled below the temperature of ignition.

The vertical direction of the current of gases in the furnace makes it difficult to secure any considerable admixture of air from the fire door, the chief dependence for the air supply must, therefore, be on the air that rises through the grates and passes upwards through the bed of fuel. These conditions make it essential for complete and economical combustion that a sufficient supply of air be admitted through the grate itself, and that the supply be well distributed over the whole grate area so that it may become mixed with the gases almost as soon as they are formed. It is also important that the grate be placed far enough below the tubes to permit of a thorough mixture of the gas and air and of complete combustion of the gas before it enters the space between the tubes. The distance from the grate to the tubes should be regulated in accordance with the volatile contents of the coal; for anthracite or coke, the minimum distance is about 24 inches; for coals containing large

quantities of volatile matter and burning with a long flame, a distance of 36 inches or more is often needed.

34. A study of the construction of the furnaces of most water-tube boilers and of the mechanical stokers generally used for burning bituminous coals will reveal the fact that a large combustion chamber, often covered by a firebrick arch for maintaining a high temperature and shielding the burning gases from the cooling effect of the surface of the boiler, is almost universally used. It is urged by some authorities that a construction that separates the hydrocarbon flame from the heating surface of the boiler is a serious disadvantage in that a large part of the heat of such a flame is given up to the boiler by radiation from the incandescent carbon particles, while a colorless gas radiates but little heat and gives up heat to the boiler almost entirely by direct contact. It is, however, certain that the carbon particles may easily part with so much heat by radiation as to be cooled below their ignition temperature, and so pass off unconsumed, with the consequent formation of black smoke; on the other hand, an arrangement of the heating surface is possible that will enable all portions of a hot current of colorless gas to be brought into sufficiently close contact with the surfaces of the boiler to give up its heat in a thoroughly effective manner. It is probable that herein lies much of the effectiveness of the heating surface of the water-tube boiler; the gas passes in thin sheets between the staggered rows of tubes or around the various baffle plates; cross-currents are thus induced that force every portion of the gas into close contact with some part of the heating surface.

ADMISSION OF STEAM INTO THE FURNACE.

35. When steam is admitted into the furnace through the ash-pit, or otherwise, it is decomposed by the heat into its constituents, hydrogen and oxygen. This process, however, absorbs as much heat as can possibly be developed by the combustion of the hydrogen thus formed. From this it follows that there can be no possible gain in heat from

introducing steam. In fact, there are several features that may cause the use of steam to result in an actual loss of heat; there is danger that much of the hydrogen liberated by the decomposition of the steam may escape unburned owing to the reduction of temperature of the fuel bed produced by the decomposition of the steam. Also, the steam enters the ash-pit at a temperature seldom above 212° F. and passes into the chimney at the temperature of the flue gases, which is rarely below 400°. It thus carries more heat into the chimney than it introduces into the furnace.

36. The use of steam or water under the grate is, however, beneficial in some cases. With coals that tend to clinker badly and to stick to the grates, the effect of the steam is to prevent the clinkering to a considerable extent; this enables the fireman to keep the grate cleaner, prevents the destruction of the grate, and, on account of the improved condition of the fire, permits of a better distribution of air and more complete and economical combustion. In the case of the argand blower and similar devices for securing an increased draft, there is often an increased economy in the use of the coal over that which is obtained by natural draft; the gain, however, must be ascribed solely to the improved air supply, and not to the fact that steam is used. A similar improvement in the draft by means of a better chimney or some mechanical draft appliance will generally give even better results than can be obtained by the use of steam. If a steam blower must be used to improve the draft, it should be so constructed and operated as to introduce as little steam into the fire as will secure the requisite air supply.

37. Rules for Economical Combustion.—Having seen that complete, i. e., economical, combustion depends on a sufficient air supply intimately mixed with the combustibles and a high furnace and combustion chamber temperature to insure ignition, the following rules will be self-evident:

1. *Fire light and often.*
2. *Keep the fire as thin as circumstances will permit.*

3. *Keep the fire clean.*

4. *Keep the space between the grate bars clear.*

5. *Keep the ash-pit clear.*

6. *When using bituminous coal, use the coking firing system, if possible.*

7. *Regulate the draft so that it will be strongest when a fresh charge of coal is put into the furnace.*

8. *Do not let the fire burn out in spots.*

9. *Do not let the fire burn too low before charging.*

10. *If possible, fire at regular intervals and in regular charges.*

HEAT LOSSES AND THEIR PREVENTION.

38. A portion of the heat generated in the furnace is usefully expended in evaporating water, but a large percentage of the heat is often wasted. Some of these heat losses are unavoidable and others are due to poor management or poor design of the boiler.

The heat losses due to moisture in the coal, incomplete combustion of the carbon and hydrocarbons, the formation of smoke, excessive air supply, etc. have already been pointed out and the methods of preventing them have been explained. In addition to these losses, there is the loss of heat inseparable from the use of natural draft. Since the draft depends entirely on the difference in density between the gases within the chimney and the air surrounding the chimney, it follows that the heat required to produce the difference in density, that is, to produce the draft, while not available for the generation of steam, is essential to the operation of a boiler plant; while the heat thus expended may be called a heat loss, it is an unavoidable loss. However, in many cases, there is a large amount of heat passing out of the chimney in excess of that required to produce the necessary intensity of draft. In practice, it has been found that when the temperature of the escaping gases has been lowered to about 500° F., the draft will be ample. If their

temperature is in excess of this, it usually indicates a serious heat loss. The high temperature may be due to several causes, either singly or combined, among which may be mentioned insufficient heating surface, inefficient heating surface, and poor water circulation.

It is rather hard to decide where to place the blame in case the temperature of the escaping gases is excessive. In general, the trouble is that the efficiency of the heating surfaces has become impaired by reason of the collection of soot and condensible tarry vapors on the fire side and the deposit of scale on the water side. The obvious remedy is to clean the surfaces and, thereafter, to clean them at such intervals as will keep them in a state of high efficiency.

39. If the heating surfaces are clean and the temperature of the escaping gases is excessive, the trouble may be due to poor circulation. It is difficult to state just exactly what should be done to improve it, since boilers vary so much in design. In horizontal return-tubular boilers, the trouble is often due to the tubes being packed too close together. Taking out one or two vertical rows has often resulted in a decided improvement in the circulation. In flue boilers and water-tube boilers, the circulation is usually free and strong; an excessive temperature of the escaping gases with these types of boilers is usually due to dirty or insufficient heating surfaces.

Insufficiency of heating surface is generally found in cases where, due to the exigencies of service, boilers are forced beyond the steam-making capacity for which they were installed. This calls for an increased combustion rate per square foot of grate surface, in consequence whereof there is an increased volume of gas that passes through the tubes and over the heating surface at a higher velocity. Since the transfer of heat from the heated gases to the water depends to a large extent on the time they are in contact with the heating surfaces, a proportionally smaller amount of heat per pound of gases is transferred to the water. There is no simple remedy for loss due to this cause. About

the only thing that can be done to make the plant more economical is to install more boilers, or perhaps a device for heating the feedwater by the waste gases may be placed into their path before they reach the chimney.

Some heat is lost by radiation from the boiler itself and some from the setting and connections. This loss, while it cannot be prevented entirely, can be minimized by covering the exposed parts with some good non-conducting material. To reduce the radiation losses of externally fired boilers, which, as a general rule, are greater than those of internally fired boilers, the brickwork setting is often made double; that is, there are two distinct walls separated by an air space. This answers very well indeed when built in such a manner that the air within the air space cannot escape.

40. The **efficiency** of a boiler plant is the ratio of the difference between the heat in the steam delivered by the boiler and the heat in the feedwater to the heat that would be developed by the perfect combustion of the fuel. This efficiency may be divided into two factors—the efficiency of the furnace as a heat producer and the efficiency of the boiler as a heat absorber. It is possible to have a furnace so well constructed and managed that the combustion is nearly perfect and yet to have a low efficiency of the plant as a whole, owing to inefficient heating surfaces, large radiation losses, etc. In order to secure a high efficiency of the plant, as a whole, it is necessary to pay strict attention to each and every detail and to keep it in the most perfect condition possible.

AUTOMATIC FURNACES AND MECHANICAL STOKERS.

INTRODUCTION.

DEVELOPMENT.

1. The earliest mechanical stoker is thought to have been invented by Watt, who obtained a patent in 1785 on a simple device for pushing the coal, after it had been coked, from the front of the grate back towards the bridge. Since that time English engineers have invented a large number of stokers, some of which have been quite extensively used and have given satisfactory results when applied to the proper conditions. None of the English designs have been much used in the United States, but a number of American designs of mechanical stokers and automatic furnaces, differing more or less from the earlier English types, have been developed since 1873, and several have been put into extensive use.

ADVANTAGES AND DISADVANTAGES.

2. Numerous tests have shown that a careful and intelligent fireman with a properly designed furnace can obtain as good results, so far as economy in the use of fuel is concerned, as have ever been obtained with any mechanical stoking device; it is also certain that hand firing may be so regulated as to produce practically smokeless combustion. It is well known, however, that these possible results are not generally attained in everyday work. Boiler firing is

hard and, in many cases, far from pleasant work. Most boiler rooms are hot and many are poorly lighted and ventilated—conditions that make it difficult for any but the best of men to keep up their interest in their work, and such men can soon demand better pay than is generally given to firemen. With the best automatic stokers the fireman is relieved from much of the most severe and difficult part of his work; he is thus more free to devote suitable care and attention to the operation of the furnace. The coal is fed to the furnace at a uniform rate and in such a manner that the gases distilled from it are thoroughly mixed with a proper supply of air; the gases are then conducted through a part of the furnace in which there is a high enough temperature to secure their complete combustion. When the coal supply and air supply are properly adjusted to suit the working conditions, the continuous and uniform manner in which the fuel is fed to the furnace insures a high and practically uniform temperature, which is favorable for the complete combustion of the gases and relieves the boiler from the stresses produced by the sudden changes in temperature that occur when cold air enters the fire door during hand firing.

ECONOMIC CONSIDERATIONS.

3. Automatic furnaces are more expensive, both in first cost and maintenance, than furnaces for hand firing, and in small plants they save but little or nothing in the cost of labor; in these cases the question of economy in their use depends on the possibility of a saving in coal and of wear and tear on the boiler. In the matter of coal they have the advantage of successfully burning cheap grades of fuel that could not be used with ordinary methods of hand firing. Automatic furnaces will give better results in the matter of smoke prevention than can be obtained by hand firing, unless an unusual degree of care and attention is given to the management of the fires. In large plants, especially where some of the modern systems of coal- and ash-handling machinery are used, automatic furnaces effect a very considerable

saving in labor; this, in addition to their other points of superiority, makes them more economical than hand firing in many cases.

GENERAL CLASSIFICATION.

4. The principal designs of mechanical stokers and automatic furnaces may be divided into two general classes, *overfeed* and *underfeed.* In the former the coal is slowly fed by some suitable mechanical device on a coking plate, where the volatile matter is distilled off by the heat of the furnace and mixed with a suitable supply of air. The coke so formed is then fed forwards on to grates, where it is burned. The mixture of gas and air is burned in a suitable combustion chamber, usually in as close proximity as is practicable to the bed of burning coke. In the second class the coal is forced by some mechanical device into a chamber *under* the mass of burning fuel in the furnace. The volatile matter is here distilled off and mixed with a supply of air. The coke formed is pushed upwards by the fresh coal that is fed into the chamber and burns above the coking chamber and on suitable grates at the sides, on which it falls. The mixture of gas and air rises through the bed of burning coke above the coking chamber and, being highly heated and thoroughly mixed, burns readily.

EXAMPLES OF MECHANICAL STOKERS.

OVERFEED STOKERS.

INCLINED OR STEPPED GRATES.

5. In the earliest American **overfeed stokers,** the fixed carbon of the coal is burned on inclined grates, consisting either of straight bars or a series of steps. The coal, after being coked on the dead plate, is pushed on to these grates, which are given a sufficiently rapid vibratory motion to feed it down at such a rate that practically all the carbon is

burned before reaching the lower end, where the ashes and clinkers are discharged.

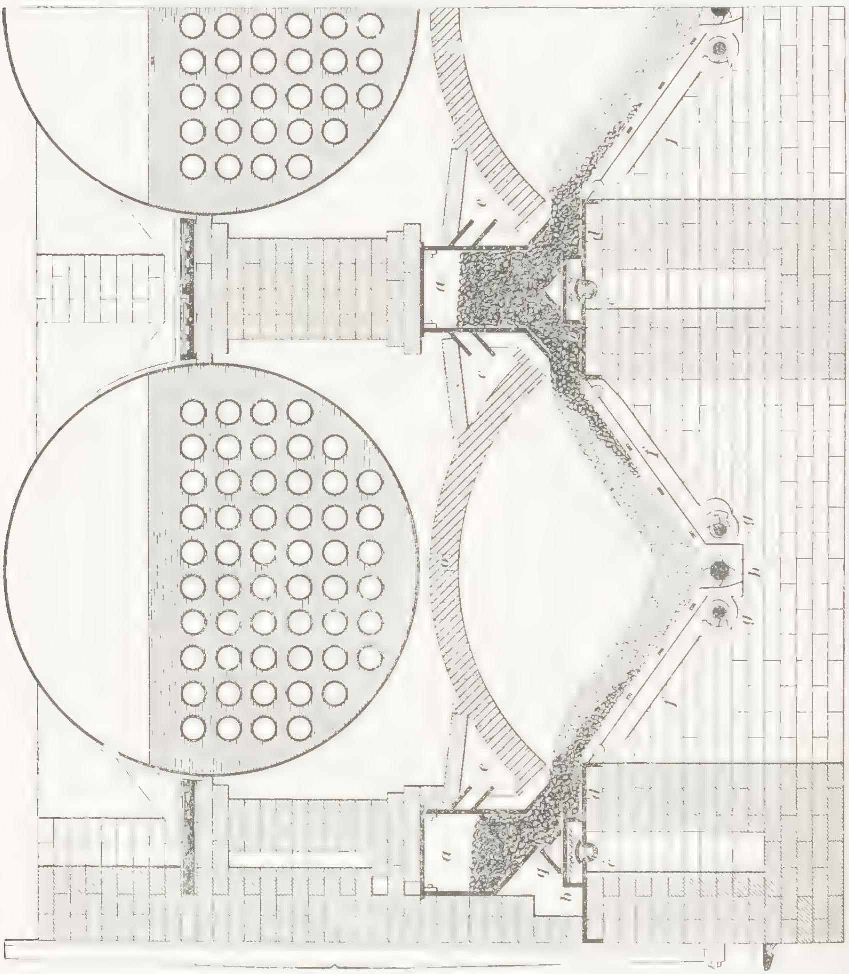


FIG. 1.

6. The **Murphy automatic furnace** is shown in Figs. 1 and 2, of which Fig. 1 is a cross-section as applied to a battery of horizontal tubular boilers, and Fig. 2 a perspective sectional view as applied to a water-tube boiler. Like parts have been lettered the same in both figures.

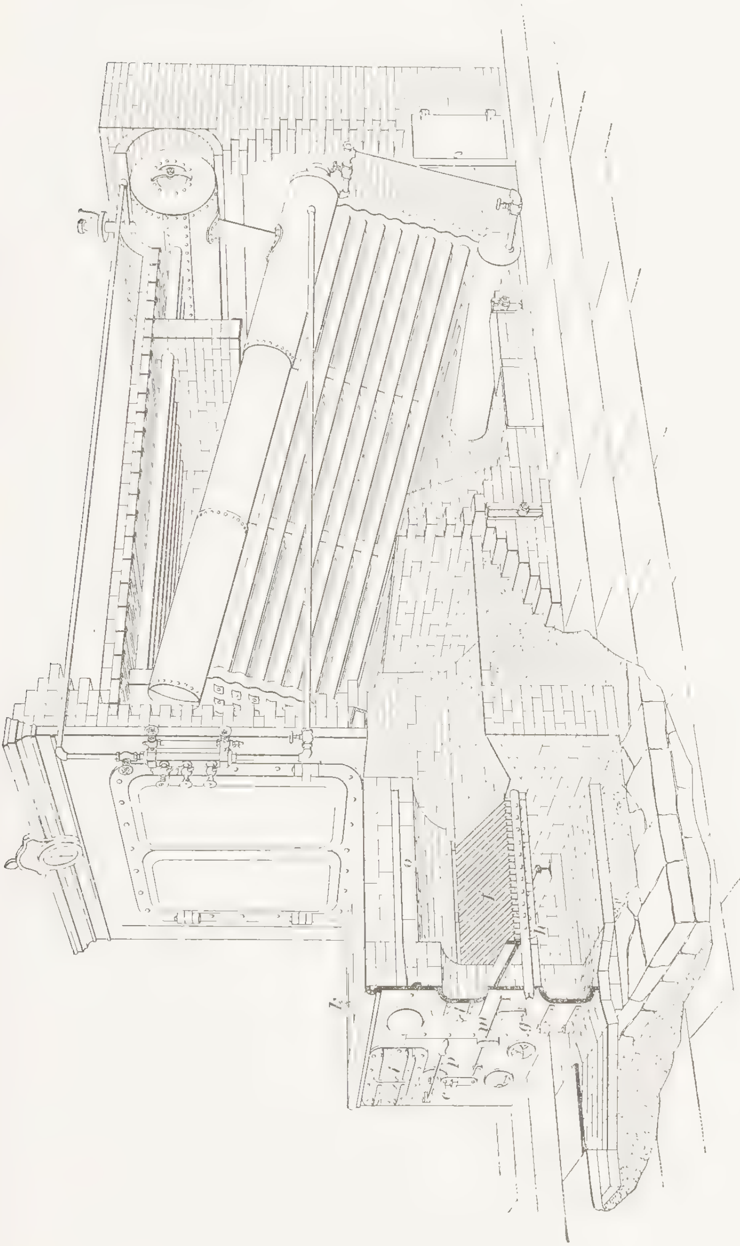


FIG. 2.

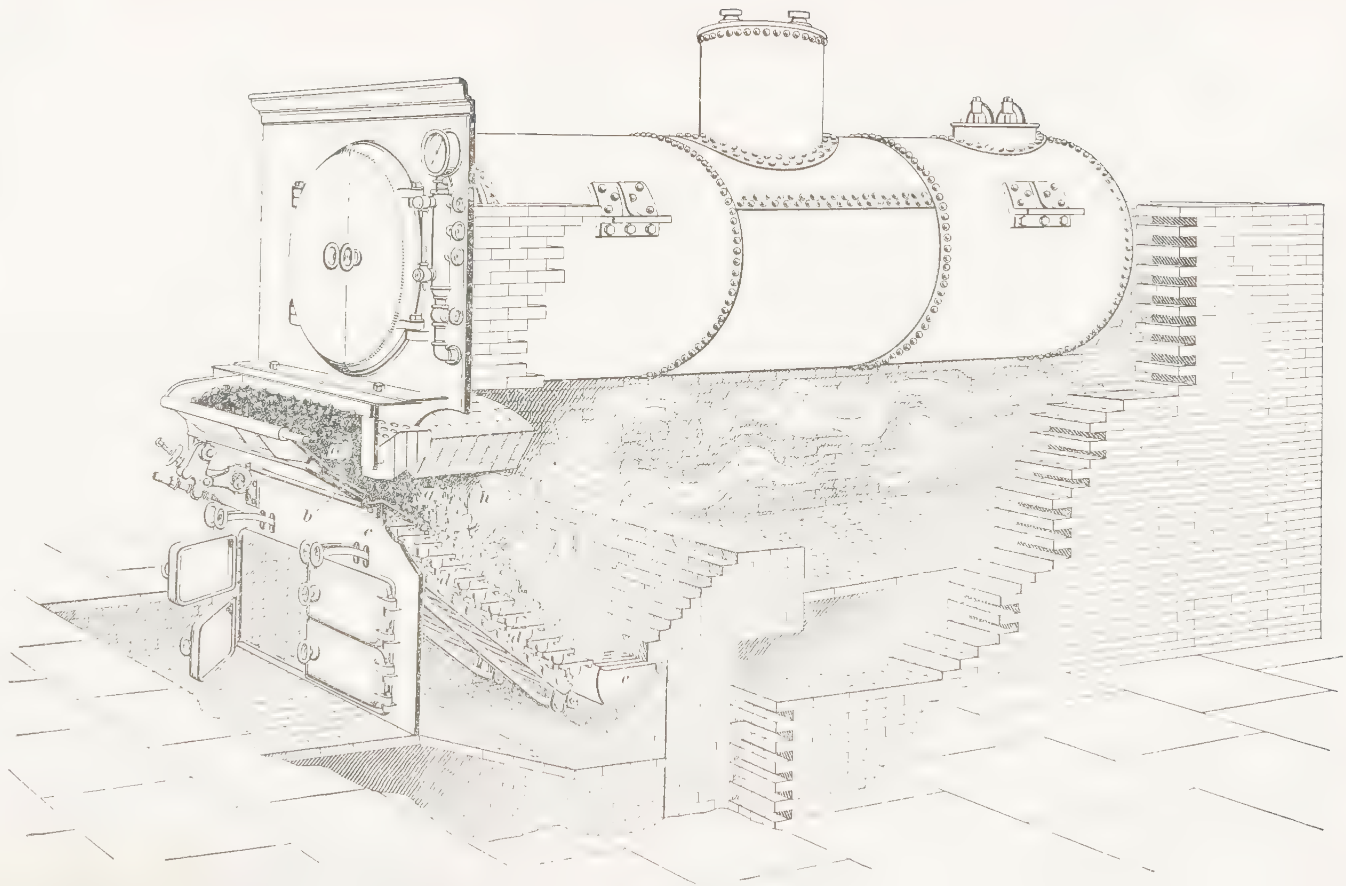


FIG. 3.

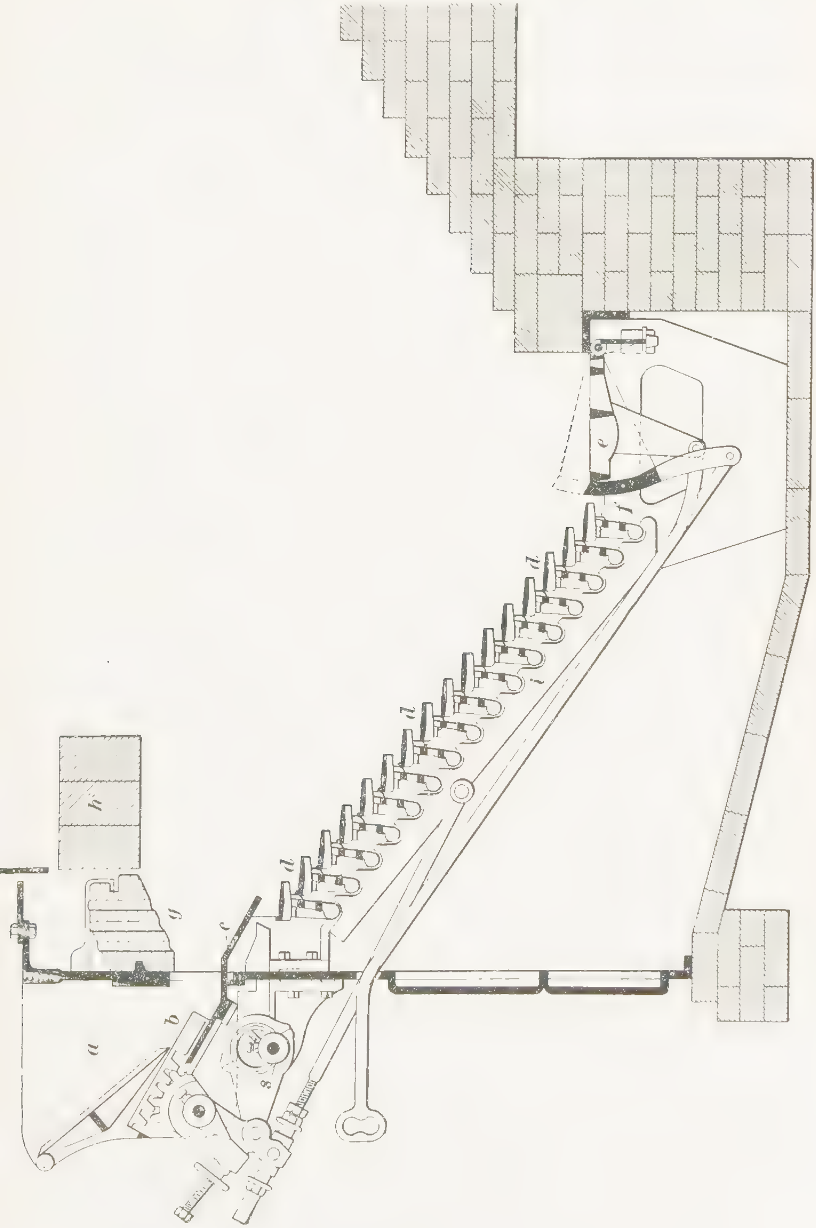


FIG. 4.

The coal, which must be in a moderately fine condition, bituminous slack being well suited for this purpose, is fed into the magazines *a*, *a*, Fig. 1, either through the doors *i* or through an opening *k*, Fig. 2. The latter method is generally employed when coal-handling machinery is used. From the magazine the coal is pushed on the coking plate *d*, Fig. 1, by the inverted box *b*, which is given a slow, reciprocating motion by the shaft *c* and a rack and pinion. The gases expelled on the coking plate mix with jets of air, which are brought in above the firebrick arch *o* through the passages *e*, *e*, Fig. 1, and enter the furnace through narrow openings just over the coking coal. The coke is gradually pushed forwards on to the inclined grates *f*, *f*. These grates are kept in motion by cams on the shafts *g*, *g*, and gradually work the coke down; the ashes and clinkers are discharged at the lower ends on to the revolving clinker breaker *h*, which grinds up the clinkers and discharges them into the ash-pit below. The firebrick arch *o* is always kept at a high temperature; the mixture of gas and air, coming between the hot arch and the incandescent bed of coke below it, is burned under the most favorable conditions. The result is that combustion is very complete, and when the furnace is properly managed, little or no smoke is formed. The shafts *c* and *g* and the clinker breaker *h* are moved through suitable connections by the bar *m*, Fig. 2, which is given a reciprocating motion by a small steam engine placed at one side of the furnace. Triangular bars *q*, Fig. 1, placed over the pushers *b*, may be worked by levers, as *p*, Fig. 2, so as to make the coal feed down more freely if for any reason one side of the furnace is receiving less than its proper amount of fuel.

The Murphy furnace is generally worked with natural draft or induced draft, but by a proper arrangement may be adapted to forced draft.

7. The **Roney mechanical stoker** belongs to the *front-feed* type, the coal being fed through a hopper at the front of the furnace. Fig. 3 is a perspective view of this stoker as applied to a horizontal tubular boiler, and Fig. 4 is a sec-

tional view, showing the relation of the different parts to one another. Like parts have been lettered the same in the two figures. The coal is fed into the hopper *a*, from which it is pushed by the pusher plate *b* on to the dead plate *c*, where it is heated and coked. From *c* the coke passes to the grate *d d d*. This grate consists of cast-iron bars, which form a series of steps; each bar is supported at its ends by trunnions and is connected by an arm to a rocker bar *i*, Fig. 4, which is slowly moved to and fro by an eccentric on the shaft *s*, so as to rock the grates back and forth between the stepped position shown and an inclination towards the back of the furnace, and the grates thus gradually move the burning coke downwards. The ashes and clinkers are discharged from the lower grate bar on to the dumping grate *e*, which can be lowered so as to drop them into the ash-pit below. A guard *f*, Fig. 4, may be raised, as shown by the dotted lines, so as to prevent coke or coal from the grate bars from falling into the ash-pit when the dumping grate is lowered. Air for burning the gases is admitted in small jets through holes in the hot **air tile** *g*, and the mixture of gas and air is burned in the hot chamber between the firebrick arch *h* and the bed of burning coke below.

The Roney stoker is designed especially for burning all grades of bituminous coal, but may be successfully used for burning fine anthracite.

8. The **Wilkinson stoker** is a front-feed stoker designed more particularly for the burning of fine anthracite coal. In Fig. 5 it is shown applied to a horizontal return-tubular boiler, while Fig. 6 shows an enlarged view of the grate itself. Like parts have been lettered the same in both figures. The grate bars *b*, *b* are cast hollow with nearly horizontal openings leading from the interior of the bars through the risers of the steps that form the upper surface of the bars; these openings are shown in the black sectional portion of the left end of the bar. Each grate bar is given a to-and-fro motion in a horizontal direction by the rock shaft *f* and links *g*, Fig. 5, the ends of the bars being supported by, and sliding on, the hollow cast-iron bearing bars *d*, *d*. A pusher *i*,

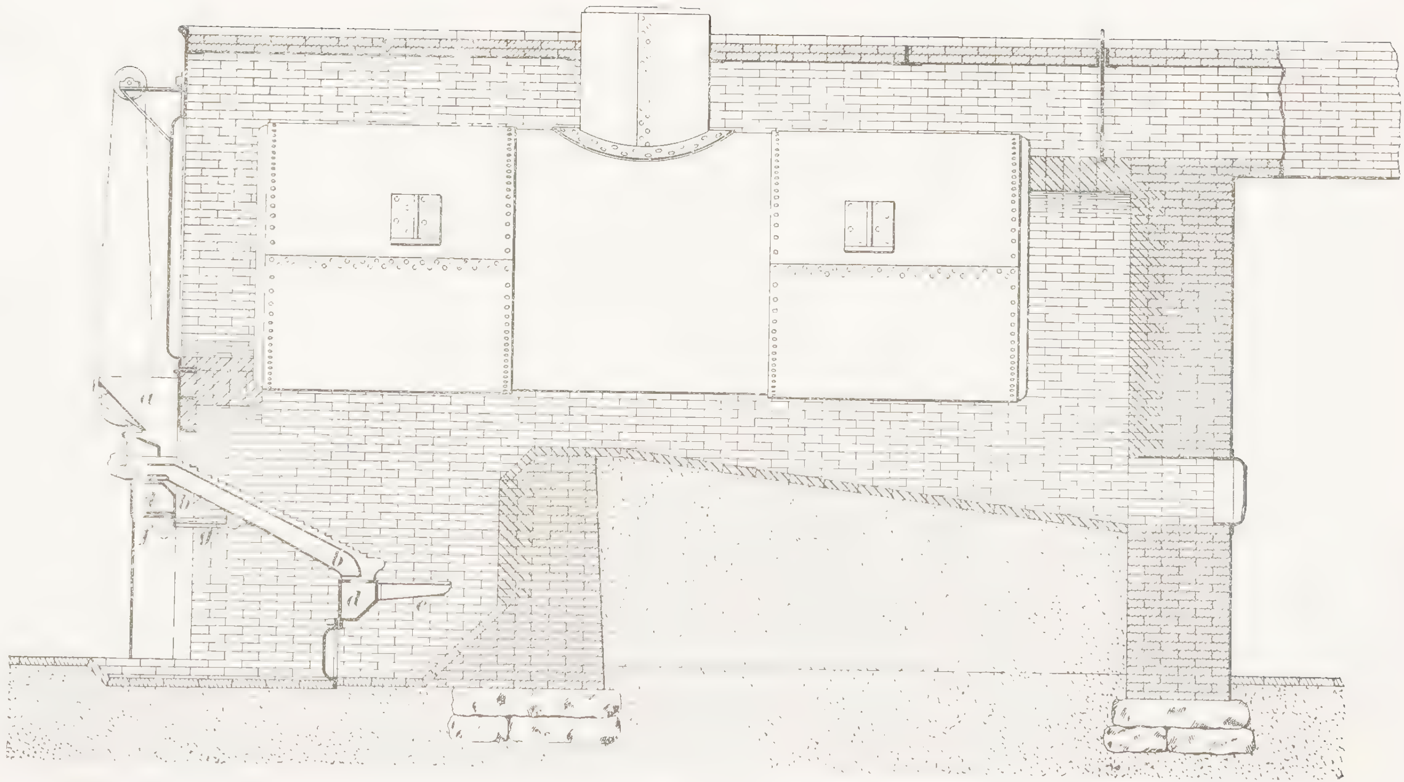


FIG. 5.

Fig. 6, fastened to the upper end of each grate bar, pushes the coal from the hopper *a* through the opening in the furnace front on the bars. The motion of the bars gradually forces the coal downwards, and deposits the ashes and clinkers on the clinker grates *e*, from which they are finally pushed into the ash-pit. Practically all the air for the combustion of the coal is drawn into the upper ends of the hollow grate bars by the steam jets *c*, Fig. 6, and forced into the fire from the openings in the tops of the bars. In this case,

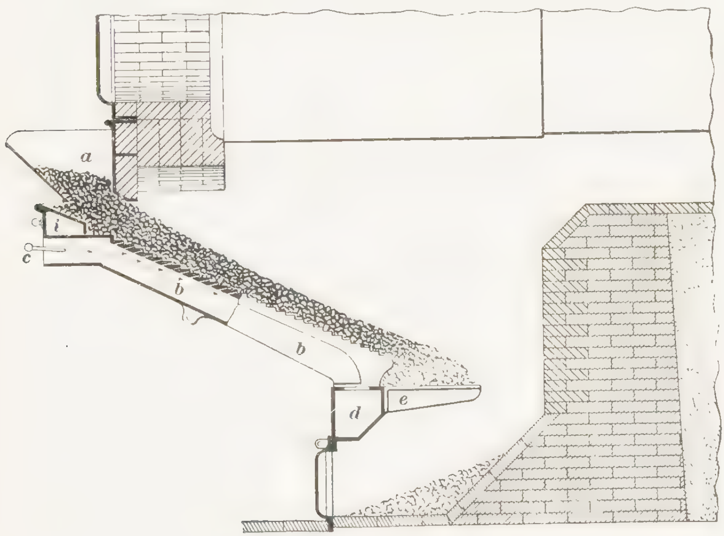


FIG. 6.

the steam jets, in addition to furnishing draft, serve an important purpose in keeping the bars moderately cool, thus preventing both their destruction by the heat and the sticking of the clinker, which with anthracite coal often causes considerable trouble if no special provision is made to overcome it. The advantages derived from this use of the steam jet are considered of sufficient importance to more than balance any possible loss of heat, and it is recommended by the makers that the steam be used, even where sufficient chimney draft is available to burn the fuel.

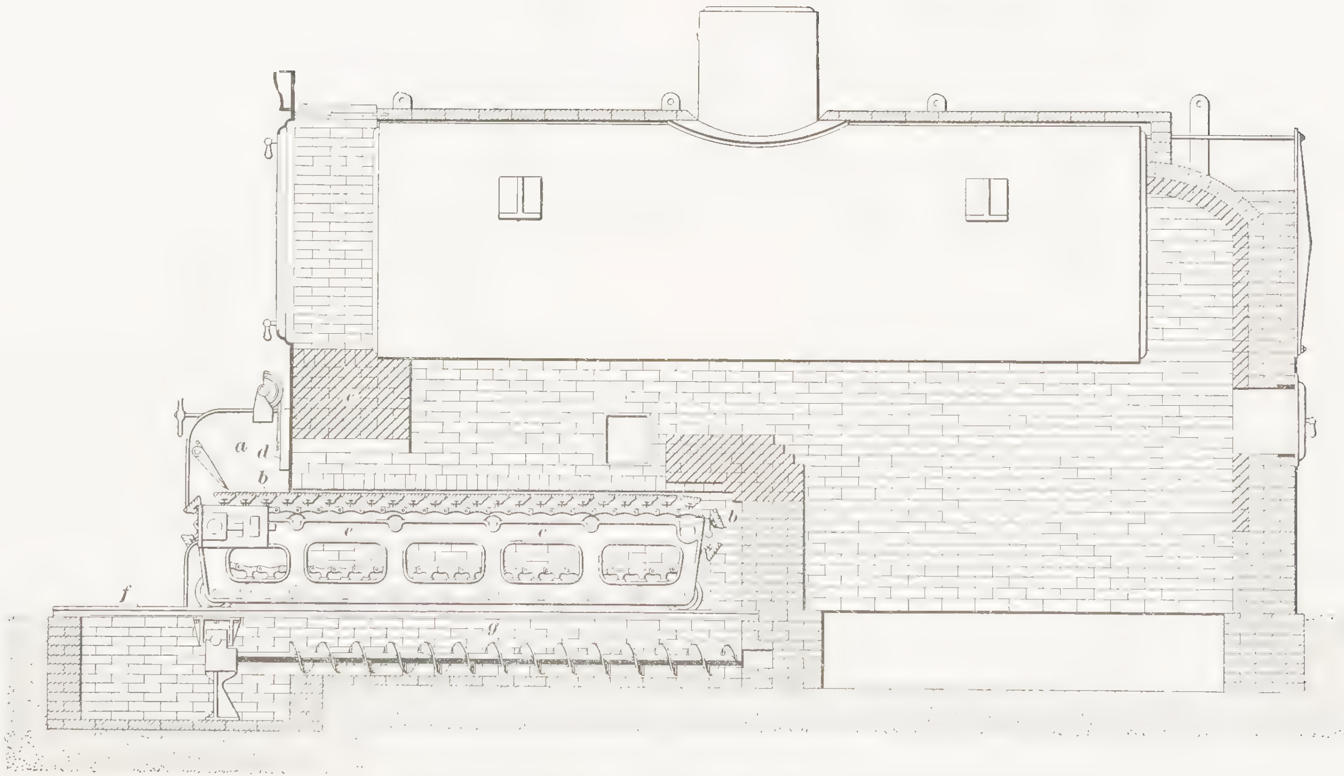


FIG. 7.

ENDLESS-CHAIN OR TRAVELING GRATES.

9. The **Playford chain-grate stoker** is shown in section in Fig. 7. It consists of a heavy cast-iron frame *e*, which is provided with suitable sprocket wheels and rollers on which travels a grate *b b* made up of sections attached to endless chains. The top of the grate is driven slowly towards the rear of the furnace, taking with it coal from the hopper *a*. The amount of coal fed to the furnace is regulated by the speed of the grate and by the opening of a gate *d*, which is water-cooled to prevent the heat of the fire from igniting the coal in the hopper. The gas is distilled from the coal in the front of the furnace under the firebrick arch *c* and burns as it rises and passes towards the back. The motion of the grate carries the coke backwards at a rate that permits the carbon to be completely burned before the rear end of the furnace is reached. The ashes and clinkers are dumped into the ash-pit at the back. A spiral conveyer *g* conveys the ashes from the rear of the furnace to a point near the front or to any convenient point from which they can be removed. The frame *e* rests on rollers that run on rails *f* and make it possible to withdraw the stoker from the furnace when repairs are needed.

In order to make the removal of burned-out grates easy and inexpensive, the grates are made in small sections, as *a*, Fig. 8, which slide over steel **T** bars *b*. The latter are in

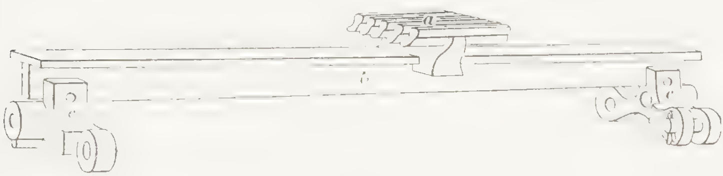


FIG. 8.

turn easily removed from the chain links *c* by taking out the pins at the ends.

10. The **Coxe mechanical stoker,** a sectional view of which is shown in Fig. 9, is a chain-grate stoker that was

designed especially for the purpose of burning the finest sizes of anthracite coal. The coal is fed through the hopper *a* and passes over firebrick "ignition blocks," which are kept at a temperature sufficiently high to insure the heating of the coal to its ignition point before it reaches the grate, where it meets an air supply. These ignition blocks are not needed when the furnace is used for bituminous coal, which burns more readily than anthracite.

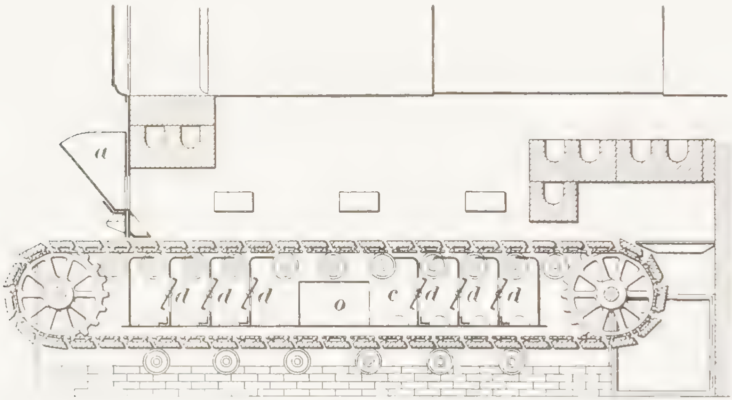


FIG. 9.

The fine anthracite coal for which this furnace was designed requires a forced draft in order to be successfully burned; and in order to regulate the draft according to the thickness of the fuel bed and the general condition of the fire, the air is conducted into the large central chamber *c* through an opening *o*. This chamber is under that part of the fire which, owing to its thickness and partly burned condition, demands the greatest intensity of draft. Openings regulated by dampers *d*, *d*, *d* conduct the air from the central chamber through a series of smaller chambers towards the ends of the furnace; by varying the opening of the dampers, the intensity of the draft in the smaller chambers can be so regulated as to give each part of the fire the amount of air needed to secure the best rate of combustion. With this furnace, the preference is given to draft produced

by a fan; the grate is so made that there is comparatively little danger from overheating, and the sticking of clinker is prevented by the traveling motion, which discharges all into the ash-pit as the grate passes over the end. Combustion is more rapid with the dry air than it would be if steam were used, and there is not as much danger of loss of heat. However, with some coals that clinker badly, it is advantageous to use a steam blower for forcing air into the space below the grate, the steam thus admitted greatly reducing the tendency to clinker.

UNDERFEED STOKERS.

11. In order to secure a high temperature of the gas and air, a number of systems of firing in which the gas liberated from the freshly fired coal, together with most of the air required for its combustion, are drawn through the bed of burning coke have been devised. Such systems, if properly managed, bring the mixture of gas and air into the closest possible contact with the incandescent coke, and, consequently, secure practically perfect and smokeless combustion. The mechanical stokers to which this principle has been applied are known as **underfeed stokers**; the coal is forced by some mechanical device into a magazine or chamber and then through an opening at the top into the bed of burning coke. In this magazine distillation takes place; the coke that is formed in the magazine is forced upwards by the fresh supplies of coal and burns above and at the sides of the magazine. The gas produced meets a supply of air from openings in the sides of the chamber and the mixture rises through the bed of burning coke.

12. The **American stoker** illustrates very clearly the principles of construction of the underfeed stoker. Fig. 10 shows sectional views of this stoker as applied to a return-tubular boiler. Coal is fed into the hopper *a*, from which it

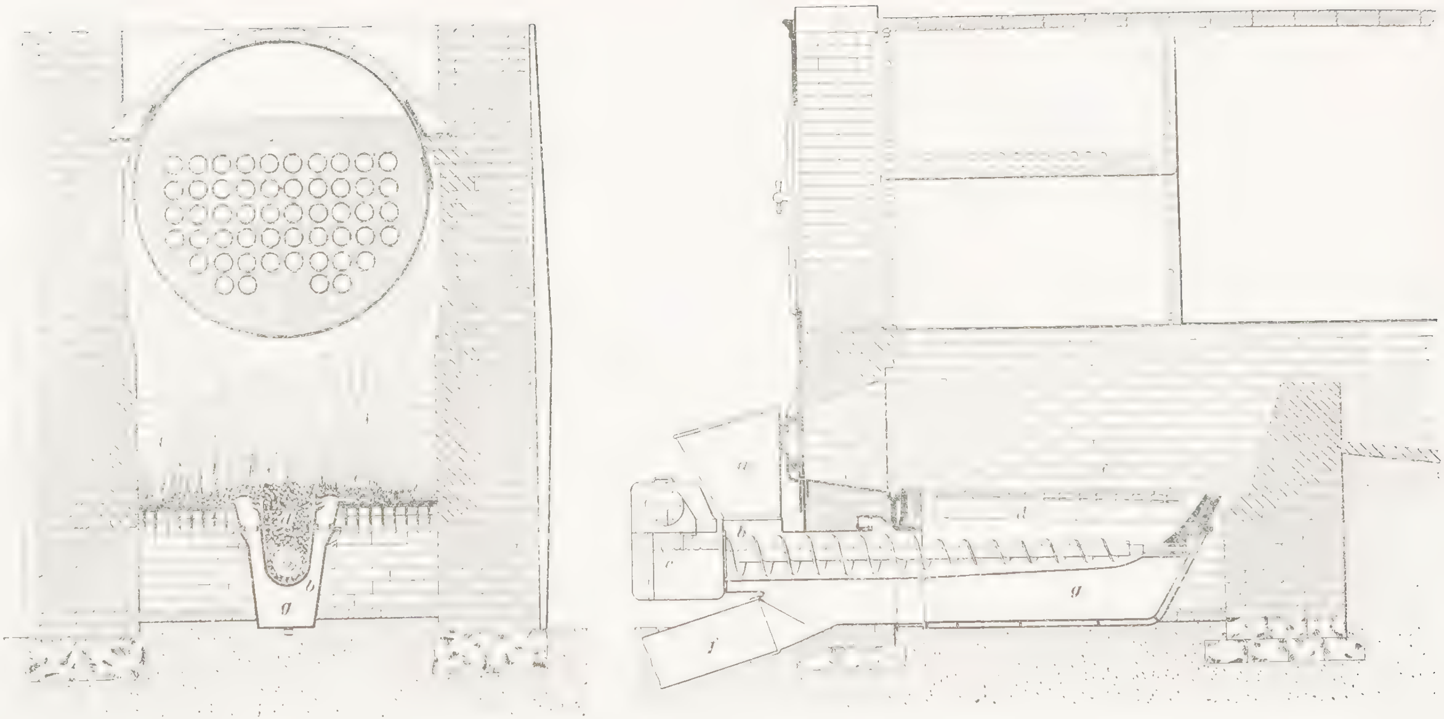


FIG. 10.

is drawn by the spiral conveyer *b* and forced into the magazine *d*, in which it is coked. The incoming supply of fresh fuel forces the coke upwards to the surface and over the sides of the magazine on the grates *i*, *i*, where it is burned. A blower forces air through a pipe *f* into the chamber *g* surrounding the magazine. From *g* the air passes upwards through the hollow cast-iron **tuyere blocks** and out through the openings, or **tuyeres,** *c*, *c*, *c*. The gas formed in the magazine, mixed with the jets of air from the tuyeres, rises through the burning coke above, where it is subjected to a sufficiently high temperature to secure its combustion. Nearly all the air for burning the coke is supplied through the tuyeres, only a very small portion of the supply coming through the grate.

The ashes and clinkers are gradually forced to the sides of the grate against the side walls of the furnace, from which they are removed from time to time through doors in the furnace front similar to the fire doors of an ordinary furnace.

13. The construction of this stoker is such that the fire must be cleaned and the ashes removed by hand. This has the disadvantage of a somewhat greater expenditure of labor than is required with those furnaces that discharge their ashes into the ash-pit, especially where it is desired to use ash-handling machinery; it also subjects the boiler to the deleterious influences of inrushes of cold air when the cleaning doors are opened. In this connection it may be stated that it is claimed by the makers that the fires do not need cleaning oftener than once in 8 or 10 hours with the poorer grades of coal, and that once in 12 hours is sufficient with the better grades; it is also a fact that all furnaces require occasional hand stirring and cleaning in order to secure a thoroughly satisfactory distribution of the fire on the grates and to prevent the formation of masses of clinkers that will occasionally stick to the grates, no matter how carefully the stoker is designed and operated.

DOWN-DRAFT FURNACES.

14. The **Hawley down-draft furnace** secures a thorough mixture of the gas and air, passes them through a mass of burning fuel, and finally burns them in a combustion chamber over another mass of burning coke. These results are obtained by the use of a furnace with two grates, one placed above the other, as shown in Fig. 11. The coal is charged through the upper doors upon the upper grate *a*, where it is coked and partially burned. Air enters the space above this grate through the upper row of doors and is drawn downwards through the grate and fuel, mixing with the gas given off from the coal. The mixture is heated in its downward passage through the fuel on the upper grate and burns in the space below; combustion in this space, which forms a sort of combustion chamber, is promoted by the heat given off by the burning coke on the lower grate *b*. The coke is but partly burned on the upper grate, the unburned portion falling through and being completely burned on the lower grate, air for this purpose being supplied through the lower doors and ash-pit in the usual manner. To prevent the rapid destruction of the upper grates by the action of the heat of the burning coal and gases, the bars are composed of inclined water tubes, expanded into headers. The front header is connected to the water space of the boiler at a point near the bottom, and the rear header is connected at a point near the surface of the water. The inclined position of the tubes and the difference in level at which the headers are connected to the water space of the boiler induces a circulation of water through the system that prevents

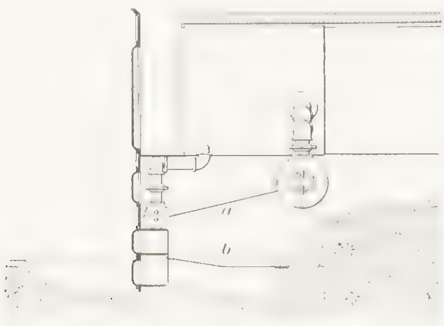


Fig. 11.

overheating. This construction also adds considerably to the heating surface of the boiler.

The Hawley furnace is a successful smoke-prevention device and the makers claim that it will burn low-grade fuels with a high efficiency. It is not automatic in its action and is therefore not as well adapted to the saving of labor in the fireroom nor for use with coal-handling machinery as most of the automatic stokers that have been described; this, however, is not a serious objection in small plants.

BOILER INSTALLATION.

BOILER SETTING.

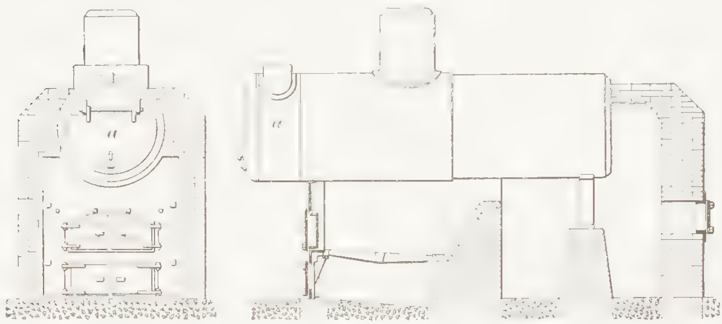
BOILER FRONTS.

1. The two styles of boiler setting in general use for tubular and flue boilers of the externally fired type are known, according to the construction of the boiler front, as the **half-arch front** setting and the **full-arch,** or **full-flush, front** setting. In the half-arch front setting, as shown in Fig. 1 (a), the smokebox a is made of metal and projects beyond the boiler front; it either forms part of the boiler itself or is separate and fastened to the front boiler head. In the full-flush front setting, the sides of the smokebox are formed by the brick setting and the front by the boiler front, as is shown in Fig. 1 (b).

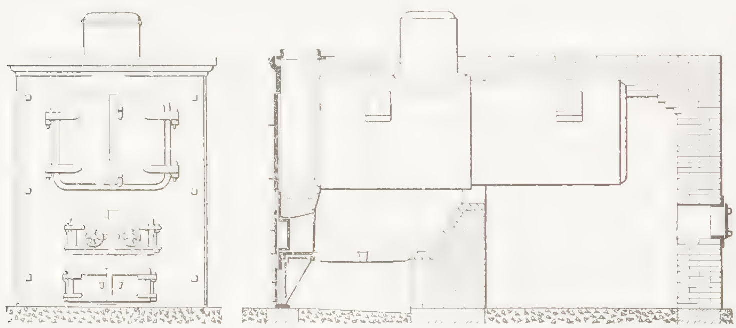
2. The half-arch front setting has the advantage that it will occupy slightly less floor space and hence will take a smaller number of common bricks and firebricks than the full-flush front setting. In general, it will be from 15 to 18 inches shorter; the width of the setting will be the same in both styles. An objection urged against the half-arch front setting is that the projecting smokebox interferes, to some extent, with the work of the fireman.

DESIGN OF BOILER SETTINGS.

3. Purpose.—In a boiler setting, three things are to be attained: (1) A firm support for the boiler shell; (2) properly arranged space for furnace and ash-pit; (3) a protective covering for the boiler that will, as far as possible, prevent loss of heat by radiation.



(a)



(b)

FIG. 1

4. Supporting Boilers.—Externally fired boilers may be supported by cast-iron lugs riveted to the shell and resting on the side walls, or they may be suspended from overhead girders by means of hooks or rings. The former method is usually adopted for the comparatively short return-tubular boiler, while the latter is used for the long

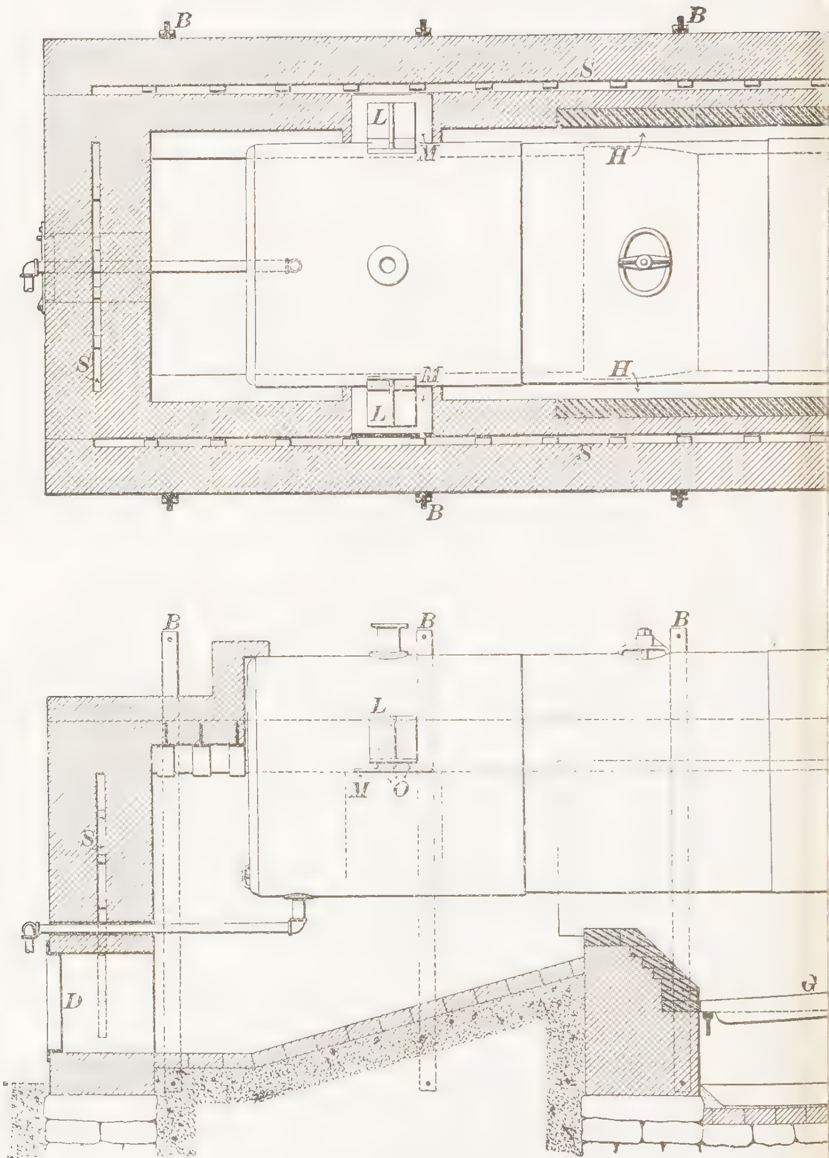
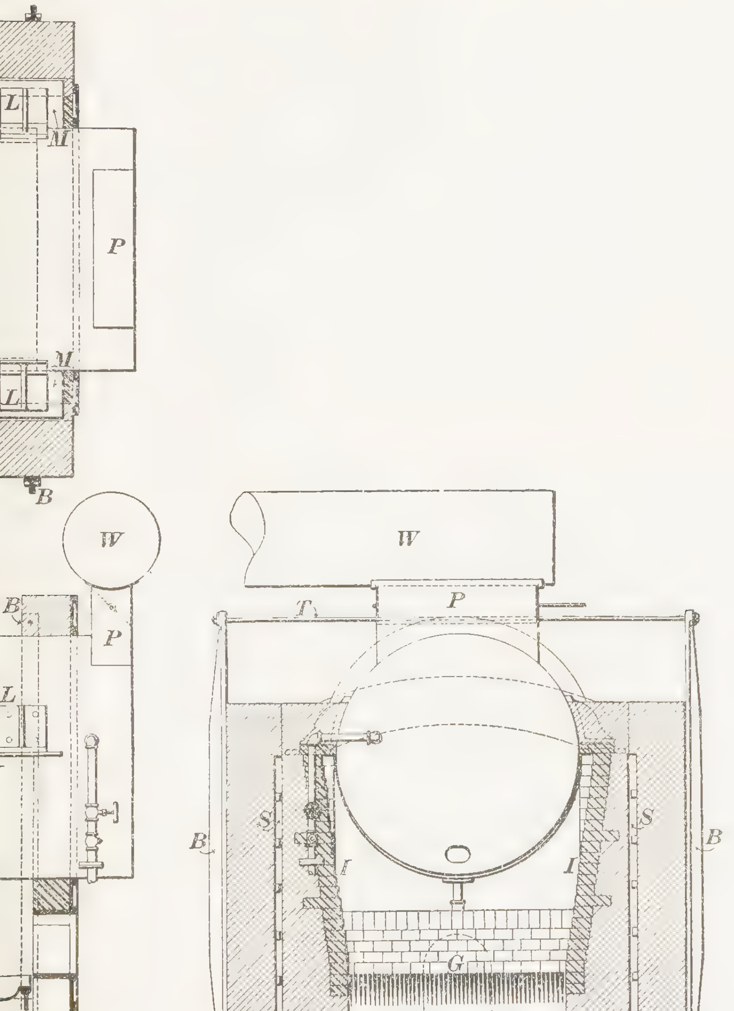


FIG.


L
M
P
M
L
B
W
B
P
L
W
P
T
S
S
B
B
I
I
G

plain cylindrical and flue boilers. When very long cylindrical boilers are suspended at two points only, the excessive weight between the supports throws a heavy stress on the lower plates in the middle of the boiler; when a center support is added, the condition of things is still worse, because the lower part of the shell expands more than the upper, which causes the shell to sag in the middle, thus throwing all the weight on the center support. Numerous cases have occurred where the center support has given way under the stress and the shell has been ruptured by the shock. It is, therefore, important in supporting these long shells to arrange the supports so that each will bear its proper proportion of the load and at the same time allow the boiler to expand freely under all conditions of temperature.

5. Example of a Good Setting.—The setting of a 60-inch return-tubular boiler with a half-arch front, as designed by the Hartford Boiler Insurance Company, is shown in Fig. 2. The foundation is heavy stonework laid to a depth of 3 or 4 feet below the surface. On top of this is laid the brickwork. The side and rear walls are double, with a 2-inch air space between the inner and outer parts. The inside wall *I*, next to the furnace, is faced with firebrick, as is also the bridge and all portions in direct contact with the flames.

The boiler is supported by cast-iron lugs *L* riveted to the shell. These lugs rest on iron plates *M* placed on the top of the side walls. The front lugs rest directly on the plates, but the back lugs rest on rollers *O* of 1-inch round iron. The boiler is thus free to expand and contract. The rear wall is 24 inches from the rear head of the boiler, so as to allow the gases an opportunity to enter the tubes; above the tubes, however, the wall is built in to meet the head and forms a roof for the chamber. The rear wall is provided with a door *D* to remove the dirt and soot that collects back of the bridge and to provide a means of inspection.

The grate G is placed 24 inches below the shell; this is a sufficient distance for anthracite coal, but for bituminous coal it might better be 30 to 36 inches. The grate has a fall of 3 inches from front to rear, which facilitates the admission of air to the rear of the grate and makes it somewhat easier to clear the spaces between the grate bars from below. The back end of the boiler should be set about 1 inch lower than the front end; this insures a thorough draining of the boiler when the blow-off is open.

The brickwork is closed into contact with the shell at the level of the center of the upper row of tubes; this prevents the gases coming in contact with the plates above the water-line. Some boilermakers prefer to make a brickwork arch over the top of the boiler and to allow the gases to pass back to the rear through the flue thus formed. The practice is risky, as it may lead to the overheating of the upper plates. A safe rule is "Never expose to fire or gases of combustion any part of the shell not completely covered with water." This rule applies to the blow-off pipe as well, which when not in use is empty; in order to prevent its destruction by the gases of combustion and the heat, it should always be protected either by covering it with a larger pipe or by a cast-iron sleeve, or by bricking it in. The last method has the serious objection that it interferes with the examination of the pipe, which may corrode badly without it being discovered when bricked in.

The brickwork is strengthened by buckstaves B held together by tie-rods T. The buckstaves are best made of wrought-iron channel or angle irons. It will be noticed that in the present case the flue pipe P is rectangular, but the pipe W leading to the chimney is cylindrical. The purpose of the air space S is to prevent the conduction of heat to the outer walls and thus keep them cool. Its utility is somewhat doubtful and many of the best boilermakers do not recommend it.

6. Setting of Cylindrical and Flue Boilers.—Plain cylindrical and flue boilers are set in about the same manner

as the return tubular. Sometimes, however, when the shells are extremely long, two or even more bridges are placed beneath the shell to keep the heated gases in contact with it.

7. Self-Contained Boilers.—Vertical and locomotive boilers and nearly all internally fired boilers are self-contained and require no setting. The vertical boiler is supported by the cast-iron base that forms the ash-pit. Firebox boilers, when stationary, are supported on cast-iron saddles and skids. It is not customary to provide vertical boilers and stationary firebox boilers with any protective covering.

8. Foundations, Firebrick Lining, and Walls.—In boiler settings the walls have not only the weight of the boiler and its attachments to sustain, but they must also resist the varying stresses caused by the alternate heating and cooling of the entire masonry. For this reason the foundations should be unusually heavy and the walls of ample thickness and properly lined with firebrick on the inside. Every sixth course of firebrick from the grate up must be a row of headers bonded into the masonry behind. By the term "**headers**" is meant that the bricks are set in with the ends as the exposed surface instead of the sides, as is the case with the other courses. This method enables the bricks between each row of headers to be renewed when necessary without having to tear down the entire wall.

9. Firebrick linings suffer most where the bed of fire comes into contact with them; the frequent impact of the fire tools against the bricks also causes them to become loosened and broken. But as the first row of headers is about 12 inches from the surface of the grate, it is safe from the contact of fire and the impact of tools. The headers also give strength to the linings. Firebrick must be set in fireclay, which should be mixed thin enough to just lay on the trowel, thus permitting the bricks to lay close to one another, the

object of the fireclay principally being to fill up the existing inequalities between the bricks. The bricks should also be dipped in water before being laid, so as not to absorb that which is in the fireclay.

10. The joints between the bricks of the outer walls should be about $\frac{1}{8}$ inch thick, of good mortar, composed of 1 part lime and 5 parts of clean sand—sea sand is not suitable for this purpose because it is not sharp enough. When building the walls, allowance should be made for the boiler expanding, so that they will not suffer unduly.

The kind and extent of the bed for the foundation depend on the nature of the earth. If the earth is firm and tenacious, trenches may be dug where the walls will stand and a bed of concrete laid, upon which good, flat stones laid in cement are placed. Joints must be broken at every course, laid so that a solid foundation will be the result. Should the earth be soft and yielding, the excavation should cover the entire area of the setting and should be filled to a good thickness with stones and concrete, upon which the foundation may be started.

11. When boilers are to be set where quicksand is found, the excavation should be deep enough to admit of a good bed of gravel being rammed home to a thickness of not less than 18 inches. Upon this a bed of stone and concrete is to be placed and finally the first course of large, flat stones is well laid in cement.

12. Settings of Water-Tube Boilers.—The setting of water-tube boilers differs from that of the horizontal tubular only in details and kind, rather than in principle. Different makes of water-tube boilers require different forms of setting, which are determined largely by the construction of the boiler, and hence no general rules can be given. Usually the manufacturers of water-tube boilers furnish drawings of the setting to the buyer; it is best, speaking generally, to follow their advice in this matter.

ARRANGEMENTS OF BOILERS AND PIPING.

STEAM PLANTS.

CONTENTS OF INSTALLATION.

13. The **installation** of a complete steam plant includes the setting of the boiler or boilers, the arrangement of the various lines of piping, and the location and arrangement of the various accessories, such as feedwater heaters, purifiers, separators, economizers, feed-pumps, injectors, etc. An elaborate plant may be fitted with economizers, mechanical stokers, coal conveyers, and ash conveyers, purifiers, and other labor-saving and fuel-saving devices. On the other hand, the plant often consists simply of boilers, chimney, and feed-pump.

GENERAL DESIGN OF PIPING.

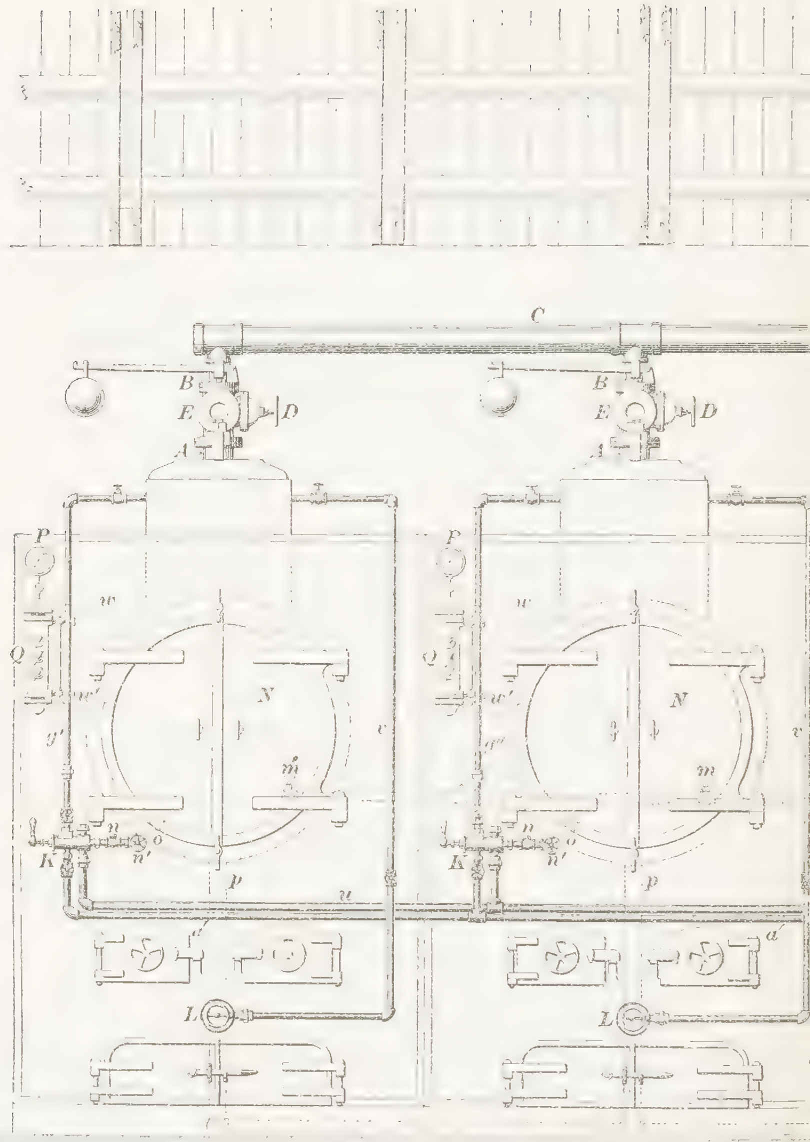
14. In arranging and designing a piping system for a steam plant, the aim must be to produce a design that combines low first cost with durability and serviceableness. Points that must be duly considered are the extent to which the piping shall be in duplicate in order to prevent a shut-down due to an accident to a section; the ease with which the piping can be taken down for repairs must also be considered. In general, flanged sections are more easily taken down than sections united by screwed joints, at least in the larger sizes. When screwed joints are used, it is advisable to introduce a liberal supply of unions to allow a section to be taken out and replaced without having to tear down the whole piping system. The question of whether to place the piping overhead or below the floor is chiefly one of convenience and looks. With the piping below the floor, the engine room will generally look better, but the piping will not be as accessible as when overhead.

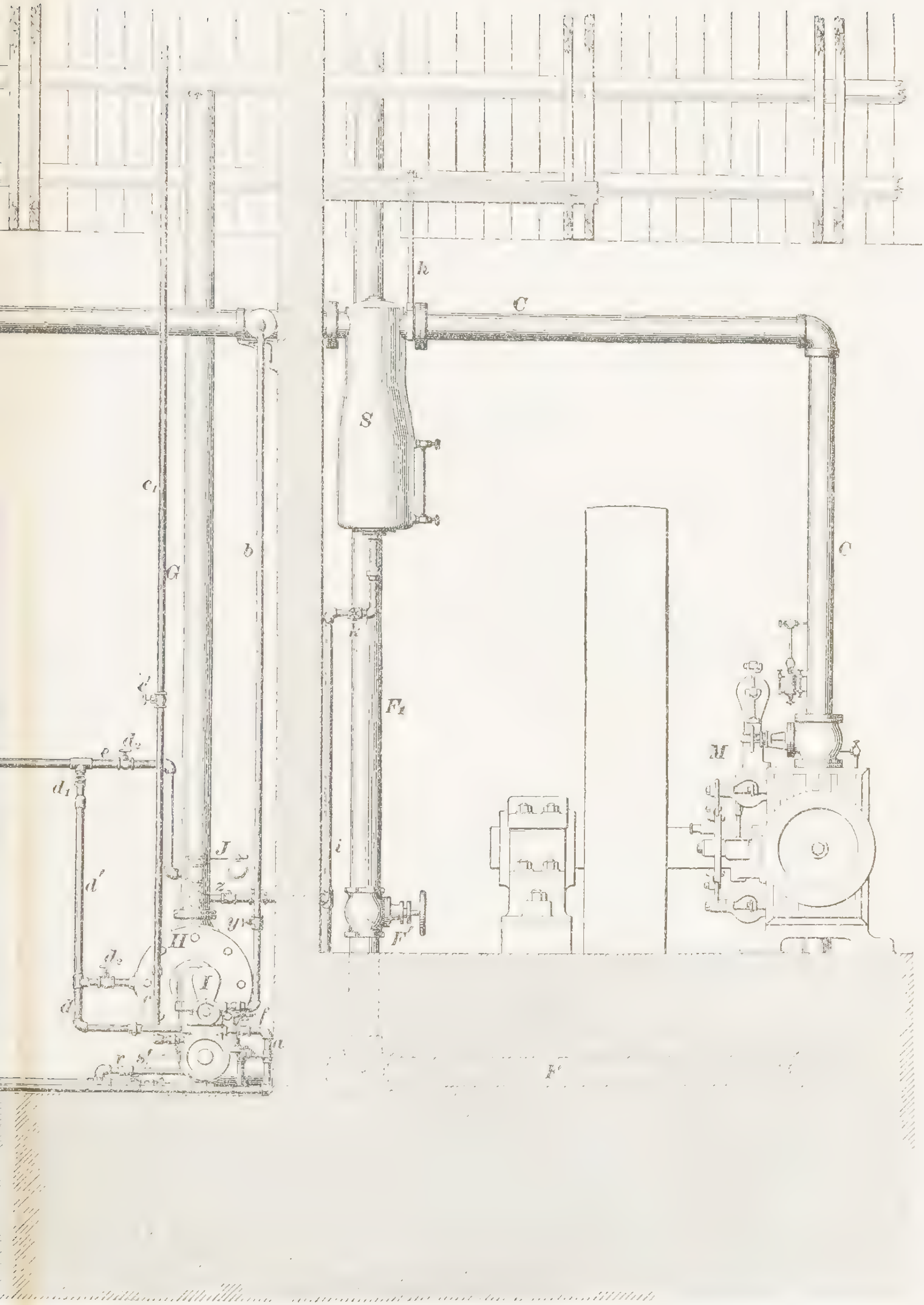
15. Example of a Plant.—There are no rules that can be given as applicable to all cases in arranging a boiler plant and the piping; each plant must be designed to suit the service for which it is intended and the local conditions. In order to give the student an idea of how a plant and the piping may be arranged, Figs. 3, 4, and 5 are given, which may be profitably studied. Fig. 3 is an elevation and Fig. 4 a top view of the plant, which has two return-tubular boilers *N*, *N* so arranged that either boiler may be used at will or both may be run at the same time.

16. In the following description the letters refer to both figures:

Suppose that the boilers have been partly filled with water, the fire started, and that the steam gauge *P* registers the desired pressure. The valves *D*, *D* are then opened and the steam is conveyed to the engine *M* through the short vertical pipes *A*, *A*, the short branch pipes *B*, *B*, and the main steam pipe *C*. It will be noticed that the safety valves *E*, *E* are attached to the upper ends of the pipes *A*, *A*, and that the valves *D*, *D* are situated between the safety valves and the main steam pipe. Hence, there is no possibility of cutting off the connection between each boiler and its safety valve.

H is a closed feedwater heater, the water being heated by the exhaust steam from the engine, which is conveyed to the heater by the exhaust pipe *F*. The water in the boiler is replenished by means of the feed-pump *I*. The pump is connected to a city reservoir, river, or other source of supply by the pipe *a*. The water is discharged through the delivery pipe *d* into the closed heater *H*, and after being heated is forced by the continued working of the pump into the boilers through the pipe *e* and its branches. The check-valves *l*, *l'* prevent the return of the water. The valves *m*, *m'* are for the purpose of shutting off the water from one of the boilers, if so desired. The valves m_1, m_2 permit either feed check-valve to be taken apart without interfering with the feeding of the other boiler. Suitable by-pass connections are provided for cutting the heater out of service for examination


C
B
E
D
A
P
Q
w
w'
y'
N
m
v
K
n
n'
o
p
u
a'
L



c 3.

and repair without interfering with the running of the plant. For this purpose a separate exhaust pipe F_1 is provided, which has a valve F'. A valve F'' is placed in the exhaust pipe F between the junction of F and F_1 and the heater. By first opening the valve F' and then closing the valve F'', the exhaust is cut off from the heater and passes through F_1 to the atmosphere. While the heater is in use, the feedwater passes through it; in order that the pump may be used while the heater is cut out, the by-pass pipe d' leads to the feed-pipe e. By opening the valve d_1 and closing the valves d_2, d_2, the feedwater is cut off from the heater and passes directly to the feedpipe e. With the heater in service, the pump exhausts into the heater through the pipe c; when the heater is cut out, the pump exhausts through the pipe c_1 to the atmosphere. To make the change, the valve c' is opened and the valve c'' closed. The feed-pump receives its steam supply from the main steam pipe C through the pipe b, and is started and stopped by operating the throttle valve y. With the heater in use, the exhaust passes to the atmosphere through the pipe G.

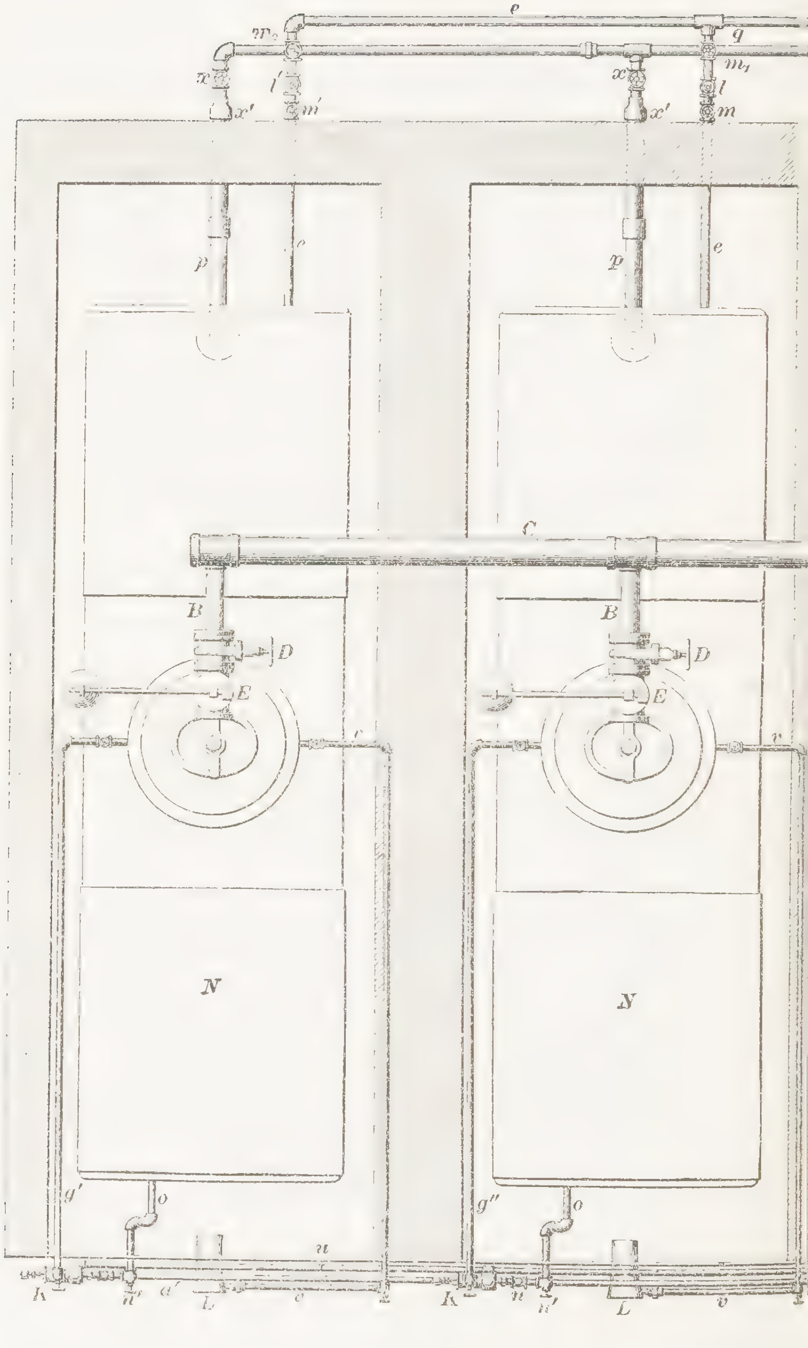
17. While in this case a separate exhaust pipe F_1 is provided for use when the heater is out of service, the same end may be attained by connecting F and G by a by-pass pipe. Thus, an elbow may be placed directly above F' and a pipe run to a **T** placed in G; in that case a valve will have to be placed in G below the **T**. This valve and the valve F'' will be closed and the valve F' opened in order to cut the heater out. A safety valve J is generally placed in the feedpipe to prevent any overpressure in the heater.

18. The blow-off pipes p, p are connected to the pipe q, which in turn is connected to the pipe t. The valves are shown at x, x. The pipes p are larger than q, for the reason that they are more likely to become choked with sediment than q. In many cases, the blow-off pipes p are independent; that is, they are not led into the same pipe. With the arrangement shown in the figure there is no way of discovering a leak in the valves x, x, should one occur. Should they

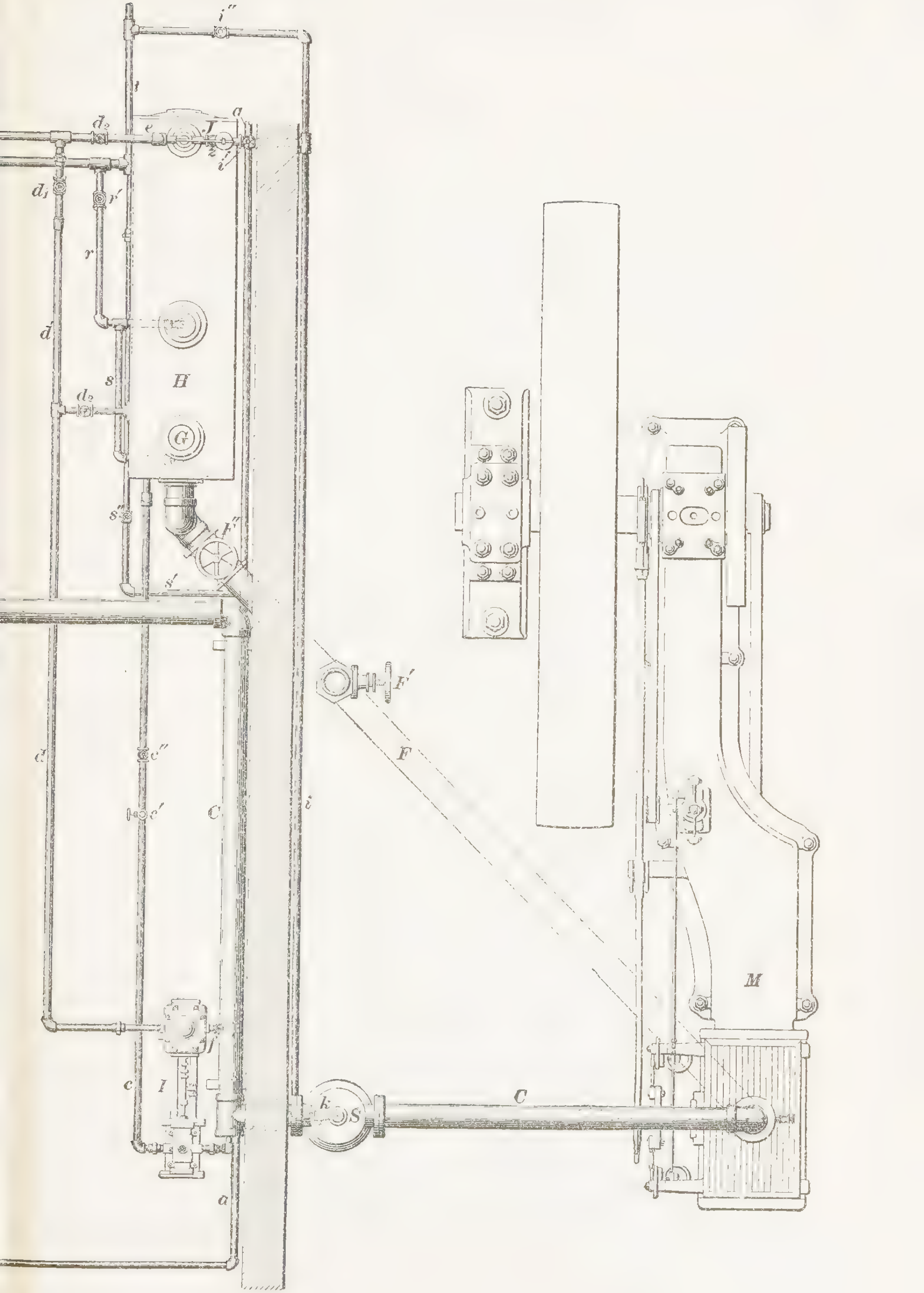
become choked, they may be readily disconnected at x' from the pipe q and the sediment removed. The pipe r is the blow-off for the heater; it connects with the main blow-off pipe q. Since the exhaust steam condenses more or less in passing through the heater, it is necessary to provide some means for getting rid of the condensed portion; this is accomplished by connecting the tubes to the blow-off pipe r by means of the pipe s. Valves are placed in r and s close to the heater; a valve r' is placed near the junction of r with q and is used for shutting off connection between the blow-off pipe and the heater drains when the heater is out of service. A small drain pipe s' is fitted to the exhaust pipe F, and connects with the blow-off pipe t. The globe valve s'' should be opened before the engine is started, so as to clear the exhaust pipe of water that may have accumulated in it. This valve should be closed again after the exhaust pipe is thoroughly warmed up and cleared of all the water.

L and L are argand blowers for producing a forced draft, the steam required being obtained from the dome by the pipes v, v.

19. In case the pump is out of order or the heater is disabled and it is considered advisable to feed hot water, the boilers can be fed by the injectors K, K. The steam for working the injectors is taken from the domes through the pipes g' and g''. The water is led to the injectors by the pipe a', which is a continuation of the pump feedpipe a and delivers the water into the boiler through the feedpipes o, o. Before starting the injectors, the valve f should be closed so as to shut off the water supply from the pump. The pipe u conducts the overflow water from the injectors to the blow-off pipe q, and a valve u' is placed in the pipe u; this valve must be closed when the boilers are being blown down in order to prevent the water backing into the overflow pipe. The check-valves n, n prevent the water escaping from the boiler through the injectors after they have stopped working. The globe valves n', n' are additional safeguards; they are for the purpose of preventing the boiler emptying



e
q
x
x'
m'
p
C
B
D
E
r
N
g'
g''
o
u
K
n
n'
L
v



G. 4.

itself after a shut-down, in case an obstruction should prevent the check-valves from closing.

20. *S* is a separator for removing the entrained water from the steam. As will be seen, it is attached to the main steam pipe *C* and is also supported by the rod *h*, which is attached to the beam overhead. The water thus removed flows down by gravity through the pipe *i* into the feedpipe *c* just below the safety valve *f* on the feedwater heater. As the temperature of the water is fully 212°, it is not necessary that it should pass through the heater. The bottom of the separator should be at least 2 feet or more above the highest water level in the boiler, since, if both were at the same level, the pump would force water into the separator and thus destroy its action. The difference of levels between the bottom of the separator and the water level in the boiler constitutes the head that induces the flow. The pipe *i* is fitted with the globe valve *k* and the check-valve *z*. The drain pipe *i* of the separator is also connected to the waste pipe *t* in order to allow the separator to be drained whenever the feedpipe *c* is cut out of service. To drain into *t*, the valve *i'* is closed and *i''* opened.

21. Endless-Piping System.—An arrangement of steam piping suitable for large power plants, in which are several units both of boilers and engines and the usual auxiliary machinery, is shown in Fig. 5. This system of piping consists of an endless main pipe divided into two sections *a*, *a'* by the main stop-valves *b*, *b'* at either end. The boilers and engines are connected to this pipe by pipes of smaller capacity, two of such pipes, as *c*, *c'*, being connected to each engine and two pipes, as *d*, *d'*, to each boiler and the two sections of the main pipe, as shown. Each section of the main steam pipe is made large enough to furnish and to deliver the required quantity of steam. Should either section of the main pipe become deranged, by the closing of the valves on the boiler pipes opening into the deranged section, it will be entirely cut off from the boilers. The valves in the pipes connecting the engines to the deranged section of pipe

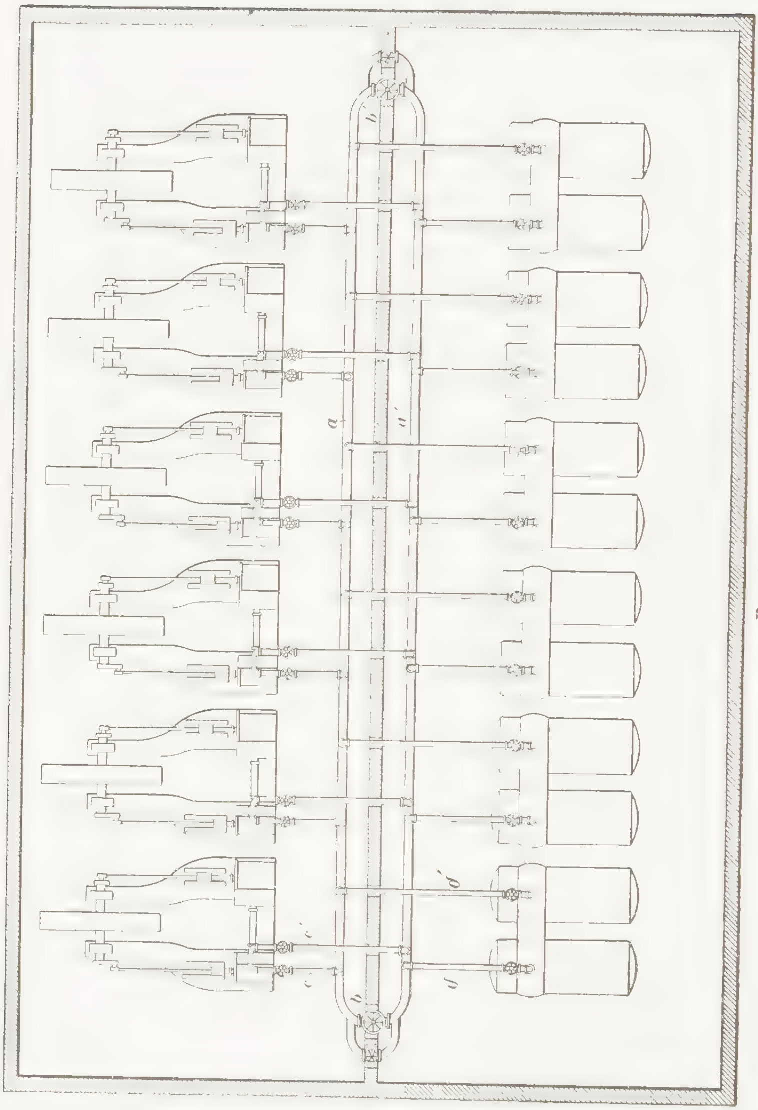


FIG. 5.

must also be closed, thus entirely cutting out the deranged section.

Each main stop-valve on the pipe has a by-pass valve used for warming up, and so also has each valve on the connecting pipes; the location of the stop-valves in the connecting pipes depends on whether the main pipe is above the boilers and engines or below the floor level. Should the main be an overhead pipe, the valves in the connecting pipes must be located close to the main and connect into the *side* of the latter, not into the bottom. The reason for this is that no condensation collecting pockets must be formed at the junction of these pipes. Regular pockets are located at different points along the bottom of the pipe for such purposes. When the main is underneath, the valves are placed at the highest point of the pipes for the reason before given. In this case the boiler-pipe stop-valves must be close to the boilers. Expansion is amply provided for in every direction in all the piping in this system by virtue of the form of the several pipes. Condensation is taken care of by a suitable system of drain pipes.

With this system of steam piping the possibility of a shut-down is very remote. As can be seen by a close study of Fig. 5, one-half the entire piping may be cut out of service through derangement or other cause and still every unit can be continued in operation by simply manipulating the proper valves. This system of piping can be installed in plants where the arrangement of boilers and engines is different from that illustrated in Fig. 5 by modifying the piping without affecting the main features.

STEAM PIPES.

INTRODUCTION.

22. Considerations.—The piping of a boiler plant requires that careful attention be paid to all the details as well as to the general design, not only in order to make it suitable for the purpose, but also in order to reduce the

liability of a breakdown. The main considerations regarding steam piping are: (1) The size of the pipes; (2) the arrangement and construction of the piping system; (3) the method of providing for expansion; (4) proper drainage.

23. Material.—The main steam pipe in the past, at least for large pipes, was made of flanged cast-iron sections bolted together; but in nearly all modern plants wrought-iron or steel piping is used. In the smaller sizes connection is made between the sections by regular pipe fittings, but in the larger sizes the ends of the pipe are threaded and screwed into cast-iron, cast-steel, or forged-iron flanges, which are then bolted together. Wrought-iron and steel pipes, generally speaking, are safer and more reliable than cast-iron pipes for steam mains and are to be preferred where high pressures are to be carried. Copper piping is used occasionally, but quite rarely.

SIZE OF PIPES.

24. Proportions According to Velocity of Flow.—Steam pipes are generally proportioned so that the velocity of flow of the steam is 6,000 feet per minute in pipes carrying live steam and 4,000 feet per minute in exhaust pipes.

On this basis the area of a steam pipe can be found by rule **1.** To find the corresponding diameter it is most convenient to use a table of the commercial sizes of pipe, choosing the nearest size; or it may be found by dividing the area by .7854 and extracting the square root of the quotient.

Rule 1.—*To find the area in square inches of a steam pipe for an engine or pump, multiply the area of the cylinder, in square inches, by the piston speed, in feet per minute, and divide the product by 6,000 for a live-steam pipe or 4,000 for an exhaust-steam pipe.*

Or, $$a = \frac{A S}{6{,}000} \text{ for live-steam pipes,}$$

and $$a = \frac{A S}{4{,}000} \text{ for exhaust pipes.}$$

where a = area of steam pipe;
A = area of cylinder;
S = piston speed.

EXAMPLE.—Find the sizes of steam and exhaust pipes for a 12″ × 24″ engine running 150 revolutions per minute.

SOLUTION.—Applying rule **1,** we get

$$a = \frac{12^2 \times .7854 \times 150 \times 2 \times \frac{24}{12}}{6,000} = 11.3 \text{ sq. in.},$$

which corresponds closely to a 4-inch pipe as the size of the steam pipe. By the same rule, we get

$$a = \frac{12^2 \times .7854 \times 150 \times 2 \times \frac{24}{12}}{4,000} = 17 \text{ sq. in., nearly},$$

as the area of the exhaust pipe. The nearest commercial size of pipe is the 5-inch. Ans.

25. Proportions According to Weight of Steam Discharged.—It sometimes happens that only the weight of the steam that is to be discharged per hour and its pressure are known. Then the area of the steam pipe can be found as follows:

Rule 2.—*Multiply the weight of steam per hour by the volume of a pound of steam at the given pressure, in cubic feet, as taken from the Steam Tables. For a live-steam pipe, multiply this product by .0004; for an exhaust-steam pipe, multiply by .0006. The product will be the area in square inches.*

Or, $a = .0004\ W\ V$ for live-steam pipes,
and $a = .0006\ W\ V$ for exhaust-steam pipes,

where a = area of steam pipe;
W = weight of steam per hour;
V = volume of a pound of steam at the given pressure.

Rule **2** is based on a constant velocity of flow of 6,000 feet per minute in live-steam pipes and 4,000 feet in exhaust-steam pipes.

EXAMPLE.—What size pipe is required to convey 3,000 pounds of live steam per hour at 85 pounds pressure?

SOLUTION.—By the Steam Tables, the volume of a pound of steam at 85 pounds gauge pressure is 4.35 cubic feet. By rule **2,**

$$a = .0004 \times 3,000 \times 4.35 = 5.22 \text{ sq. in.}$$

The corresponding diameter is $2\frac{5}{8}$ inches, nearly, and the nearest commercial size of pipe is $2\frac{1}{2}$ inches. Most engineers would prefer to use a 3-inch pipe, as the $2\frac{1}{2}$-inch pipe has an area smaller than 5.22 sq. in. Ans.

26. Sizes of Branch Pipes.—When several boilers are connected to the same main, the main does not need to be of the same size throughout. Let the boilers be numbered from 1 up, commencing with the boiler farthest from the engine. Then the section between boilers 1 and 2 need only be large enough to carry the steam from boiler number 1. The section between boilers 2 and 3 should have twice the sectional area of the first section; the portion between boilers 3 and 4 should have 3 times the sectional area of the first section, and so on. Thus, if the diameter of the first section of the main pipe is 5 inches, the diameter of the next section should be $5\sqrt{2} = 7$ inches; of the third section $5\sqrt{3} = 8\frac{1}{2}$ inches, nearly; of the fourth section $5\sqrt{4} = 10$ inches, for the reason that the areas vary as the square of the diameter.

PROVISION FOR EXPANSION.

27. Introduction.—In putting up a steam pipe, due provision must be made for expansion and contraction, which ordinarily amounts to about $1\frac{1}{2}$ inches per hundred feet of length in steam pipes. Expansion and contraction may be provided for either by a proper arrangement of the steam pipe itself or by so-called expansion joints.

28. Pipe Arrangements.—When there is but one boiler and the engine is located fairly close to the boiler, a vertical pipe is generally run up from the engine to the level of the dome and a straight pipe then runs to the dome. If the vertical pipe is not too short, it is springy enough to bend easily and all expansion and contraction is easily taken care of by it without an undue straining of the joints.

29. When several boilers discharge into the same main, the arrangement shown in Fig. 6 may be used for connecting each boiler to the main, which itself may be arranged as

shown in Figs. 3 and 4. From the dome, or from the shell, if there is no dome, the short length of pipe *A* rises vertically and connects with a horizontal branch *B*, which is joined to the main pipe *C* by the short vertical pipe *A'*. The main steam pipe is thus allowed to expand or contract freely. The only effect of the expansion is to slightly turn the pipe *B* on *A* as a pivot. The elbow joining *B* and *A'* should have a drain pipe and valve, as shown.

The branch pipe leading from the steam dome of each boiler to the steam main must, of course, be fitted with a stop-valve, as shown, in order that each boiler may be cut into or out of service.

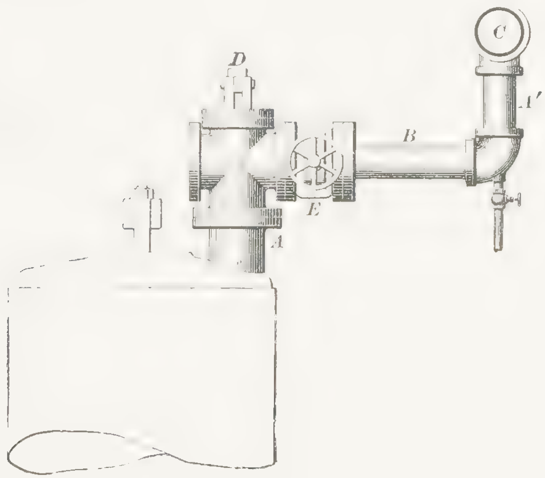


FIG. 6.

Sometimes there are two stop-valves fitted alongside of each other in the steam pipe leading from the boiler to the main and in cases where more than one boiler discharges into the main. The object of this arrangement is to facilitate operations when the boilers are subjected to a hydrostatic test. Inspectors almost invariably will refuse to subject a boiler of a battery to a hydrostatic test while the other boilers are under steam and the boiler to be tested is shut off from the steam main only by a valve. They generally require

the boiler to be entirely disconnected and to have its outlet stopped by a blank flange. Obviously this disconnecting requires the whole plant to be shut down while it is going on. But if two valves are fitted in the branch pipe and the boiler is cut out by closing both of them, most inspectors will not refuse to make the hydrostatic test, considering this arrangement to be equivalent to a disconnected boiler stopped up by a blank flange. Obviously there is no necessity of shutting the plant down in order to prepare for the test of one boiler, and hence the test can be made at any convenient time.

30. Expansion Joints.—Expansion joints may be made in various ways, the most common forms being shown in

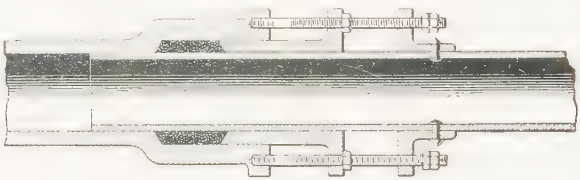


FIG. 7.

Figs. 7, 8, 9, and 10. Fig. 7 shows the **slip joint,** which is provided with a stuffingbox in order to make a steam-tight joint. Slip joints should always have a fixed flange through which studs bearing check-nuts are passed, the object being to prevent the steam pressure forcing the joint apart. The nuts are not intended to ordinarily be in contact with the flange, as that would partially or entirely destroy the action of the joint. They are to be so adjusted that the joint can never come entirely apart.

31. Fig. 8 shows a corrugated expansion joint, which is largely used on large exhaust pipes. As is shown, it is a short section of corrugated pipe, usually copper, to

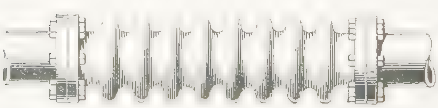


FIG. 8.

which flanges are brazed and which is put in the steam pipe in any convenient location.

32. Another way of providing for expansion and contraction is to use a length of curved copper pipe *C*, Fig. 9. The curved form of the copper section enables it to take up all the expansion or contraction of the steam pipe and also its own expansion or contraction. The ends of the copper pipe are brazed to brass flanges, which are bolted to the flanges of the cast-iron sections. The sections of pipe are bolted together at the flanges.

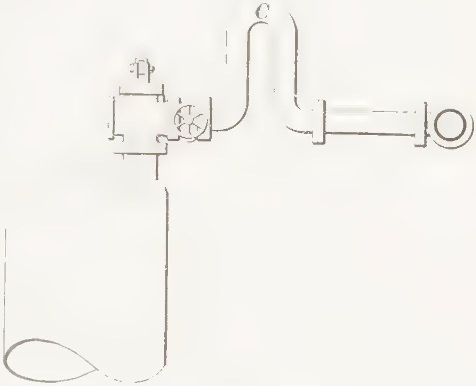


FIG. 9.

33. A slightly different form of a curved expansion pipe, which is familiarly known as a **gooseneck,** is shown in Fig. 10 (*a*). This acts in the same manner as the curved pipe shown in Fig. 9. Attention is called to the fact that curved pipes, as those in Figs. 9 and 10 (*a*), should always be placed with the bend upwards, in order that no water may collect in the bend. Fig. 10 (*b*) shows a form of bent pipe that is easily produced. While not as flexible as a gooseneck, it will still answer for many places.

34. Packing the Joints.—The joint between flanged sections of piping and between piping and expansion joints is made tight by facing the flanges and interposing between them a ring of India rubber or gutta percha. Sometimes a corrugated copper plate is placed between the flanges, and again a groove is cut in them and a copper wire placed in it

DRAINAGE.

35. Drainage is best provided for by so arranging the piping that all the water of condensation will, by gravity, flow towards a point close to the delivery end of the pipe and then

providing a drain pipe at that point. In the case of large pipes, a trap may be placed at the end of the drain pipe for automatic draining; the trap serves to seal the end of the drain pipe and thus prevents the waste of steam. In small pipes a valve is generally placed in the drain pipe, which is kept open while the steam pipe is being warmed up.

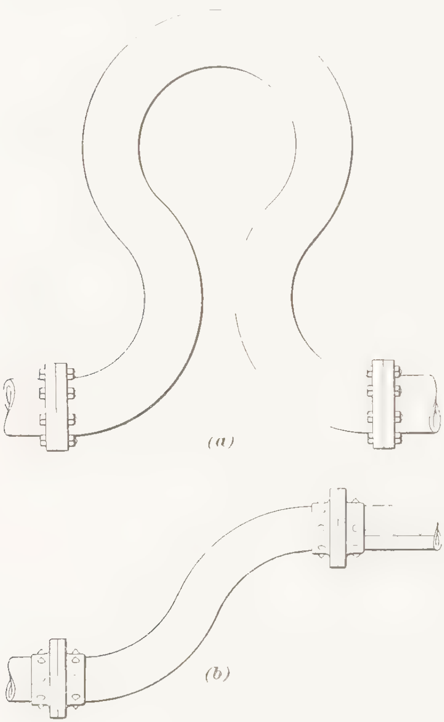


Fig. 10.

36. Steam pipes should be so arranged that no pockets or angles are formed in which water may collect. Where this cannot be avoided, the bend or angle should be provided with a drain cock or trap.

37. The presence of water in a steam pipe is the cause of the so-called "water hammer," which is so often heard in steam-heating plants. Professor Thurston has experimentally shown that the pressure produced by water hammer may be ten times that which the pipe was expected to sustain in its regular work. In some cases the water hammer has caused boiler explosions by bursting the steam pipes.

STEAM PIPE AND BOILER COVERING.

38. The tops of boilers and other portions of the surface not in contact with the furnace gases should be covered with some non-conducting substance to prevent the

radiation of heat. The same is true of the steam pipes leading from the boiler to the engine. A common and effective covering for the tops of boilers is ashes or loam. Where several boilers are set in a row, the ashes or loam is filled into the space between them until the surface is level. This form of covering is open to the objection that the ashes tend to gather moisture and hasten external corrosion, especially if the shell leaks under the covering. Another method of covering the exposed surfaces is to plaster them with a mortar of $\frac{1}{3}$ plaster of Paris and $\frac{2}{3}$ sawdust and afterwards cover them with a coating of asbestos, hair felt, or mineral wool, which may be tied on with wire. The whole is then covered with roofing paper. The following covering for steam pipes is taken from "Steam": "First wrap the pipe in asbestos paper; then lay strips of wood lengthwise along the pipe, from 6 to 12 in number, according to the size of the pipe, and bind them with wire or cord. Around this framework wrap roofing paper and fasten it with paste or twine. If exposed to weather, use tarred paper or paint the exterior." Where flanges occur, space may be left to give access to the bolts and afterwards filled up with hair felt.

39. Coverings are now manufactured by some concerns to fit the various sizes of steam pipe; they may be readily fastened to the pipe and are easily removed. They are generally made of magnesia or asbestos, or a combination of the two, and appear to serve their purpose effectively.

40. It has been demonstrated by actual experiment that the poorest covering that can be applied to steam pipes is much better (in preventing condensation) than no covering at all, so that it would be a good investment even to buy the poorest that can be obtained, where the alternative is an uncovered pipe. As a general rule, the lighter (by weight) the covering, the better it is; the less condensation it gives, the higher its efficiency. But efficiency is not the only thing to be considered when determining upon a selection. A covering must be permanent and able to resist the

temperature of the steam within the pipes for a reasonable period of time. Cork has come into use in recent years for pipe coverings and has proved very successful. It is one of the best non-conductors of heat, while at the same time it is of a durable nature, clean, and light.

41. Table I gives the results of tests made by Mr. C. L. Norton on different materials used for pipe covering.

TABLE I.

TESTS OF PIPE COVERINGS.

Name.	At 200 Pounds Steam Pressure.			
	B. T. U. loss per Sq. Ft. Pipe Surface per Minute.	Per Cent. or Ratio of Loss to Loss From Bare Pipe.	Thickness. Inches.	Weight per Foot of Length. 4 Inches Diameter. Ounces.
Nonpareil cork, double-thick	1.00	7.2	2.00	63
Nonpareil cork, standard	2.20	15.9	1.00	27
Nonpareil cork, octagonal	2.38	17.2	.80	16
Manville high pressure	2.38	17.2	1.25	54
Magnesia	2.45	17.7	1.12	35
Imperial asbestos	2.49	18.0	1.12	45
"W. B."	2.62	18.9	1.12	59
Asbestos air cell	2.77	20.0	1.12	35
Manville infusorial earth	2.80	20.2	1.50	.
Manville low pressure	2.87	20.7	1.25	..
Manville magnesia asbestos	2.88	20.8	1.50	65
Magnabestos	2.91	21.0	1.12	48
Molded sectional	3.00	21.7	1.12	41
Asbestos fire board	3.33	24.1	1.12	35
Calcite	3.61	26.1	1.12	66
Bare pipe	13.84	100.0	..	..

42. Table II shows the relative values of a number of non-conducting materials as established by tests made by Mr. C. E. Emery.

TABLE II.

RELATIVE VALUES OF NON-CONDUCTING MATERIALS.

Material.	Value.
Hair felt....................	100.0
Mineral wool No. 2............	83.2
Mineral wool No. 2 and tar....	71.5
Sawdust.......................	68.0
Mineral wool No. 1............	67.6
Charcoal......................	63.2
Pine wood across grain........	55.3
Loam..........................	55.0
Slaked lime...................	48.0
Gas-house carbon..............	47.0
Asbestos......................	36.3
Coal ashes....................	34.5
Coke in lumps.................	27.7
Air space 2 inches deep.......	13.6

STEAM APPLIANCES.

INTRODUCTION.

43. The steam appliances that aid in producing a greater economy in the use of fuel and steam that are in most common use are the *steam separator*, the *steam loop*, the *steam trap*, the *reducing valve*, the *economizer*, and the *recording gauge*. These will be described in the order named.

SEPARATORS.

PURPOSE OF SEPARATORS.

44. A **separator** is an apparatus designed to remove the entrained water, or the oil, dirt, and other impurities from a current of steam flowing through a pipe. When the

separator is intended to free the steam from water simply, it is placed on the main pipe leading from the boiler to the engine and as close as possible to the latter. When it is desired to remove the grease and dirt from the exhaust steam before condensing it and feeding it back to the boiler, the separator is placed in the exhaust pipe leading from the engine to the condenser.

CLASSIFICATION.

45. Steam separators may be divided into two general classes: (1) baffle-plate separators; (2) centrifugal separators. In a baffle-plate separator the steam comes into contact with plates placed generally at right angles to its direction of flow, thus changing the direction of flow abruptly. In a centrifugal separator the steam in flowing through the device is given a whirling (rotary) motion. In either case the action of the separator depends on inertia.

EXPLANATION OF ACTION.

46. The manner in which inertia effects a separation of the steam and water or oil may be explained thus: Since water or oil is many times heavier than the steam, the inertia of the water or oil in a current of steam is much greater than that of the steam. Consequently, when the current of steam comes into contact with a baffle plate which abruptly changes its direction, the steam changes its direction with ease, but the heavier particles of water or oil, by reason of their inertia, tend to proceed in their original direction and are dashed against the baffle plate, trickling down on it. In a centrifugal separator, the centrifugal force developed in the particles of water and oil, by reason of their greater weight, is greater than that of the particles of steam, and, consequently, they are thrown against the sides of the vessel, where they adhere and down which they trickle.

47. The chief problem in designing a separator is to prevent the separated water or oil from being mixed again with the steam after separation from the steam; the many ways in which this problem may be solved quite satisfactorily account for the large number of separators in the market.

CENTRIFUGAL SEPARATORS.

48. The **Stratton separator,** shown in Fig. 11, belongs to the centrifugal type. It consists of a chamber with a steam inlet and outlet, and containing a vertical pipe *a*. The steam enters by the inlet *c* and is deflected by a curved partition that gives it a spiral motion about the pipe *a*. The particles of water are thrown off by centrifugal action and run down the walls to the bottom of the chamber. The steam passes through the pipe *a* and out the outlet *d* in a practically dry condition. The separator is provided with a drain pipe *h* for the removal of the water and a gauge glass *g*. The wings *b*, *b* are four in number and are for the purpose of destroying the centrifugal effect of the steam after it has reached the bottom of the separator. They likewise offer additional surface for the water particles to adhere to. Were it not for these wings, the steam would keep up its rotative motion while passing up the pipe *a* and thus necessarily carry some of the entrained water along with it.

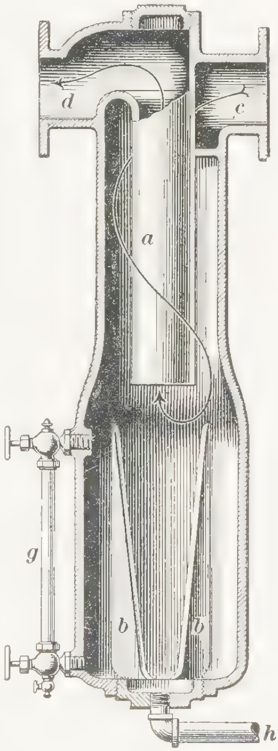


FIG. 11.

49. The **Mosher centrifugal separator** is shown in Fig. 12. In this separator the steam enters at *a* and flows

out at *b*. It is constrained to follow a helical path by the stationary worm *c*, and the centrifugal force generated

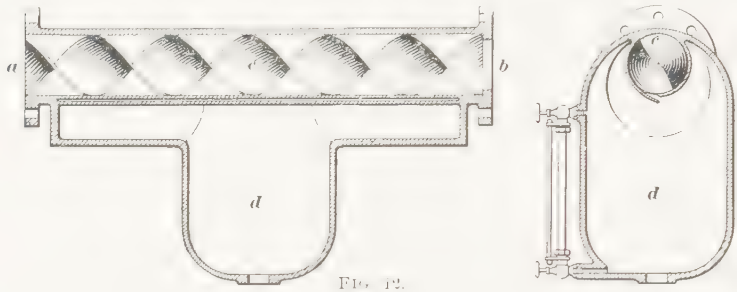


FIG. 12.

by this whirling motion of the current of steam causes the entrained water to be thrown against the sides of the chamber containing the worm, down which they flow through a long slit in the bottom into the receiver *d*.

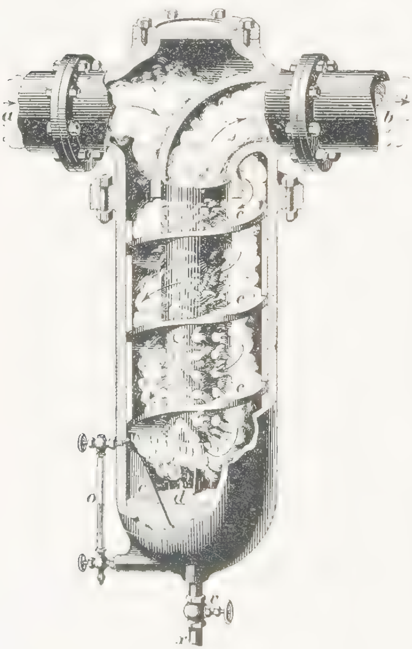


FIG. 13.

50. Fig. 13 is a sectional view of the **Simpson centrifugal steam separator.** In this the wet steam enters at *a* and is given a rapid whirling motion by the stationary helical guides *c*, *c*, first passing downwards into the bottom *d* of the separator, where its whirling motion is destroyed by wings, as *e*. The steam, which has now been freed from the entrained water, passes up the central pipe and out at *b*. The path of the steam through the separator is clearly indicated by the arrows. A gauge glass *o* shows the amount of water.

BAFFLE-PLATE SEPARATORS.

51. The **Hine steam separator,** or **eliminator,** as it is called by the makers, which is shown in Fig. 14, is a representative design of the baffle-plate class. In this separator the steam enters at *a* and strikes against the baffle plate *b*, placed at right angles to its direction of flow. The water

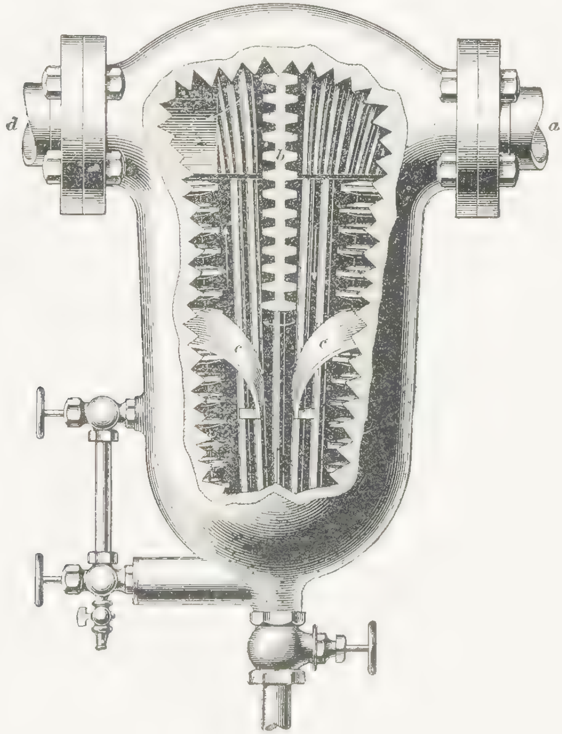


FIG. 14

or oil dashed against the baffle plate flows along the corrugations shown to the corrugated sides of the chamber and down to the bottom, whence it is removed by a periodic use of the blow-off. The curved partitions *c*, *c* assist in preventing the steam from picking up any water in the bottom. The steam leaves the separator at *d*.

52. The **Austin steam separator** shown in Fig. 15 also belongs to the baffle-plate type and greatly resembles the Hine eliminator. The steam enters at *a*, and is deflected downwards by the baffle plate *b*, which is grooved. The water dashed against the plate flows along the grooves to the sides and then to the bottom *d* of the separator. The steam after passing to the bottom again suddenly changes its direction, and any water not deposited against the baffle plate strikes against the curved partition in the form of a grate, as shown in the illustration, and falls to the bottom. The steam leaves the separator at *c*. A drain pipe *x* and stop-valve *v* are provided for draining the separator, and a glass water gauge *o* shows the amount of water in the bottom chamber.

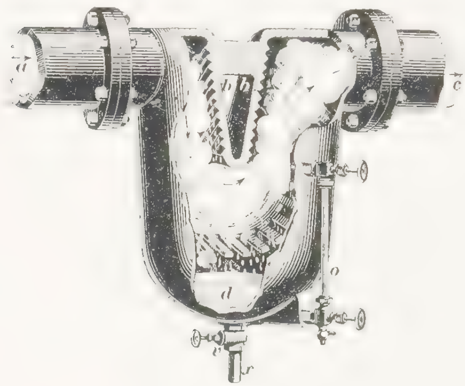


Fig. 15.

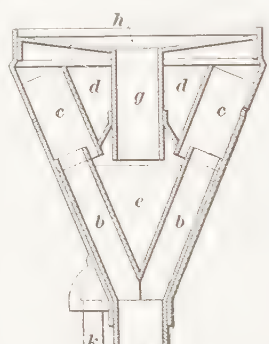


Fig. 16.

EXHAUST HEADS.

53. The **exhaust head** is a special form of separator that is to be placed on the end of the exhaust pipe of a non-condensing engine or pump in order to catch the water of condensation and prevent its being scattered promiscuously over roofs, etc. in the vicinity of the exhaust outlet. Exhaust heads, generally, also serve as **mufflers,** i. e., they deaden the sound of the exhaust. They are made in a variety of ways; all of them, however, depend on

changes in the direction of the current of steam for the separation of the water from the steam.

54. Fig. 16 shows the construction of an exhaust head in common use and serves to emphasize its similarity to the separator. The exhaust steam enters through the pipe *a*, passes up through the branches *b*, *b* and nozzles *c*, *c*, and strikes against the top head *h*. It then passes downwards through the cone *d*, and after making a sharp turn at the bottom of the cone escapes through *g*. The water lodging on the different surfaces drips into the collecting chamber and drains out through the pipe *k*.

THE STEAM LOOP.

55. The steam loop is an appliance for automatically returning the water of condensation from a steam pipe, steam-heating system, steam separator, etc. to the boiler. Fig. 17 shows its construction when applied to a separator. When its principle of operation is understood, the loop can easily be modified to suit different conditions.

56. The loop consists essentially of a **riser** *d*, a **bend** *i* acting as a check, a so-called **horizontal** *c*, a **drop leg** *f*, and a check-valve and globe valve in the pipe connecting the bottom of the drop leg to the boiler. While it is possible to operate the steam loop without a check-valve in the delivery pipe of the boiler, this should never be done on account of the danger of the water in the boiler *a* blowing back through the loop in case there should be much less than the boiler pressure in the system from which the loop is draining the water of condensation.

57. With a check-valve in the delivery pipe, the operation is as follows: Owing to the condensation of the steam, the pressure in the horizontal *c* will be slightly less than in the separator. In consequence, there will be a flow of steam up the riser *d* and through the bend *i* into the horizontal, and thence into the drop leg. Any water collected in the

separator will be carried along, and as the bend *i* prevents its return down the riser, this water flows along the horizontal pipe into the drop leg. The water continues to rise in the drop leg until the pressure due to its head being added to the pressure in the horizontal is sufficient to overcome the pressure within the boiler, when the check-valve is forced

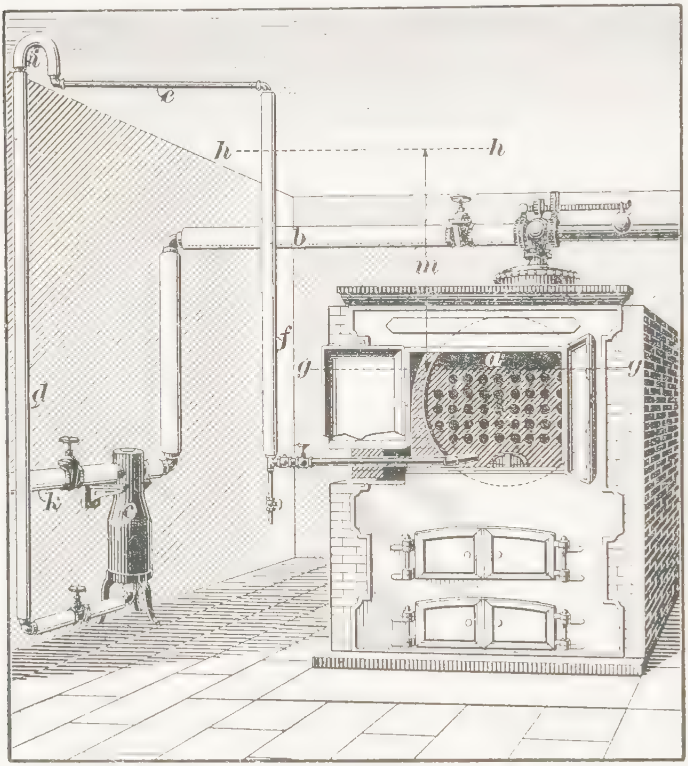


Fig. 17.

open and the water flows into the boiler until the water has dropped so low in the drop leg that the pressure in the drop leg due to the head of water and the steam pressure is less than the boiler pressure, when the check-valve closes. The head of water is measured between the water level *g g* of the boiler and the level at which the water stands in the drop

leg. Thus, if the water stands at the level *h h*, the head is given by the distance *m*.

58. The distance *m*, in feet, from which the height of the drop leg can be determined depends on the difference in pressure existing at the separator and the boiler pressure. In practice, about 2.5 feet should be allowed for each pound difference in pressures.

STEAM TRAPS.

PURPOSE.

59. A steam trap is an appliance for removing, periodically, the water of condensation from steam pipes, steam-heating systems, separators, and similar apparatus without the waste of steam. Steam traps may or may not be required on gravity-return systems (similar to gravity-feed apparatus), depending on conditions. When the boiler pressure is low, a sufficient head of water to overcome the boiler pressure can often be provided for in the vertical height of the pipe through which the water of condensation returns to the boiler; in other cases, where the boiler pressure is too high, a special steam trap may be needed above the boiler level. In many cases the water of condensation is simply discharged into a reservoir, if it is desired to save it, or into the sewer. In the latter case, a simple form of steam trap is placed on the end of the return pipe, which trap permits the free passage of water, but prohibits the discharge of steam.

CLASSIFICATION.

60. Steam traps may be divided into two general classes in accordance with their construction and purpose. These classes are *open traps* and *closed traps*.

61. An **open trap** may be defined as a trap that is constructed in such a manner that it can only discharge into a vessel in which there is a lower pressure than in the trap. An open trap can be made to discharge into a vessel having

a higher pressure than there is in the trap in one case only, which exists where the trap can be placed high enough above the vessel into which it is to discharge to have a head of water in its discharge pipe sufficiently great to create a pressure that, added to the pressure in the trap, will give a pressure higher than exists in the vessel into which it is to discharge. This condition exists in steam plants where the steam mains are overhead and are much higher than the level of the boilers; the condensation from the mains can in that case often be returned to the boilers by an open trap.

62. A **closed trap** is constructed in such a manner that it can discharge water of condensation from a system in which there is a low pressure into a vessel in which there is a high pressure. When thus used, it must be placed several feet above the vessel subjected to the high pressure. It may also be used to discharge into a vessel having a lower pressure than exists in the system it drains or into the atmosphere. It is more economical, however, to use the cheaper open trap for this purpose, generally speaking.

OPEN TRAPS.

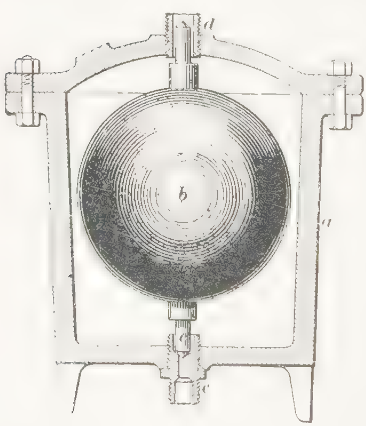


FIG. 18.

63. Float Traps.—The **Eureka steam trap,** shown in Fig. 18, is one of the simplest open traps and belongs to the **float** type. It consists of a chamber *a* containing a hollow copper float *b* having a valve *c* at its lower end and a guide at its upper end. The water of condensation enters the top of the trap through the pipe *d* and leaves the trap through a pipe attached to the nipple *e*, which contains the valve seat for the valve *c*.

The operation is as follows: With the trap empty, the weight of the float holds the valve *c* to its seat. As water commences to flow into the trap, its buoyant effect tends to raise the float, and when enough water has run in the trap, the float rises, thus opening the outlet at *c* and allowing some water to escape. As soon as sufficient water has run out of the trap to allow the weight of the float to overbalance the buoyant effect of the water, the outlet *c* is closed again by the valve *c* and the operation is ready to be repeated. The action of the trap is thus seen to be automatic and intermittent.

64. Bucket Traps.—The **Albany steam trap,** shown in Fig. 19, is an example of a **bucket** trap. The water of condensation flows through the pipe *b* into the trap body *a*. As the water rises in the trap, it floats the empty bucket *c*, and the valve shown in the center of the pot bottom closes the opening of the tube *i*, which, through the passage *e*, connects with the outlet pipe *d*.

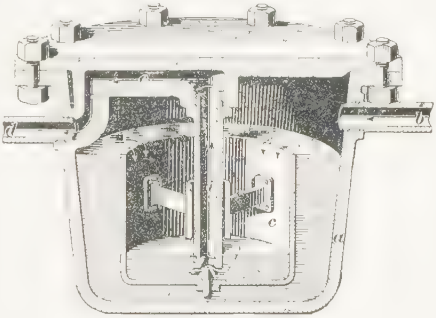


Fig. 19.

It will be understood that the trap blows steam out of the outlet only when first put into service; after the trap has been used once, there is sufficient water left in the body to hold the outlet valve closed. As the water continues to rise in the trap body, it finally overflows into the bucket until the weight of the bucket is sufficient to cause it to drop. This opens the base of the pipe *i* and the steam pressure then forces the water in the bucket out of the outlet until the buoyant effect of the water outside the bucket is large enough to raise the bucket again and thus close the valve. The operation is now ready to be repeated.

65. Expansion Traps.—There are a number of different traps in the market in which the discharge valve is operated by the expansion or contraction of some part due to changes in the temperature. While there is no doubt that they will work satisfactorily if properly taken care of, they are usually rather delicate in construction and require to be regulated very nicely, and hence will not stand as much abuse as those based on other principles.

CLOSED TRAPS.

66. Closed traps are often called **return traps,** being used chiefly for returning water of condensation to the boilers. The distinguishing feature of all return traps is that they are provided with a live-steam connection and an automatic valve operated by the float or other device controlling the emptying of the trap, which valve admits live steam to the trap to equalize the pressure within it and the boiler into which it drains.

67. An example of a return trap is given in Fig. 20, which represents the **Curtis return trap.** It consists of a receiver *a* containing the float *i* and having connected to it the water inlet pipe fitted with the check-valve *d*, the live-steam inlet pipe fitted with the stop-valve *b*, and the water outlet pipe fitted with the check-valve *c*. A rotary valve *e* serves to admit live steam to the trap. It is connected by the rod *g* to the rocker *f* by a stud working in a curved slot. The rocker *f* is engaged by a pin in the lever *h*, which

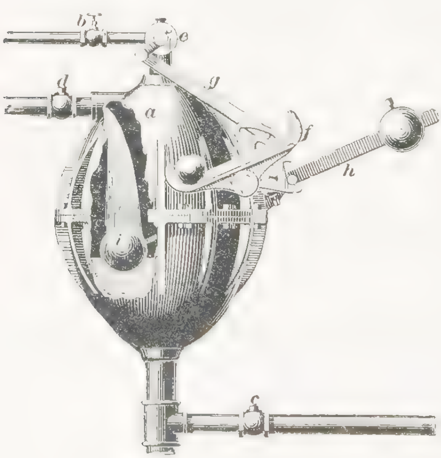


Fig. 20.

carries the float *i* on one end and a counterweight on the other.

68. The operation is as follows: At first the live steam is shut off. As the water of condensation flows into the receiver, the float *i* rises and the loaded end of *h* descends. The stud in *h* engaging a lug of the rocker *f*, the latter rotates right-handed until the track confining the ball shown on *f* is inclined to the right. The ball now rolls down the track and causes the rocker *f* to rotate to its limit, which pulls the valve *c* open and admits live steam to the trap. The receiver now empties, and as the float *i* descends, the rocker *f* will finally be rotated left-handed, which closes the steam valve. The operation is now ready to be repeated. The check-valve *d* prevents the live steam blowing back into the system being drained by the trap; the check-valve *c* prevents the water in the boiler backing up into the trap.

69. Most return traps will lift the water a reasonable distance, a vacuum being created in them through the condensation of the steam.

REDUCING VALVES.

70. The primary purpose of a reducing valve is to reduce steam from a higher to a lower pressure for various purposes. Reducing valves are used chiefly in connection with steam-heating systems, but some may be and are used for supplying steam at a *constant* pressure to engines.

71. The reducing valves in most common use are designed to maintain a *uniform* pressure at the outlet, regardless of the original pressure. The **Foster reducing valve,** shown in Fig. 21, is a representative design of this class.

The steam inlet is at *A*; the outlet may be at either *B* or *C*. Two valves *e* and *g*, connected together by the sleeve *H*, are rigidly attached to the valve stem *G*. The valves are guided by four wings, as shown; the valve stem is guided in a hole in the bonnet *U*. When the valve is not

in use, the valves c and g are wide open. The operation of the valve is as follows: When steam is first admitted, it passes through the annular openings between the valves and their seats to the outlet. Some of the steam on the outlet side of the valve passes through the small steam port E into the diaphragm chamber D; it acts on the under side of the diaphragm F and forces it upwards until the force exerted on the diaphragm by the steam balances the resistance of the spring J. As is well known, the resistance of a spring increases very rapidly when compressed. To counteract this increased resistance, the diaphragm acts on the spring through the intervention of the togglejoint $a\,a$, which allows the diaphragm to act with an increasing leverage when forced upwards. The togglejoint is so proportioned that the movement of the diaphragm is directly proportional to the steam pressure. In order that a small movement of the diaphragm may cause a large movement of the valves, the valve stem is not connected to the diaphragm, but to a pair of multiplying levers x, x. When the pressure of the entering steam is increased, the increased pressure will force the diaphragm upwards, the valves move towards their seats, and the area of opening is decreased. This wiredraws the steam to the pressure for which the valve is set. When the pressure of the entering steam is reduced, the reverse of the above takes place. Since the area of opening (and hence the steam pressure on the outlet side) depends on the relative position of the valves in regard to their seats, and since the position of the valves is governed by the resistance of the spring J, it follows that

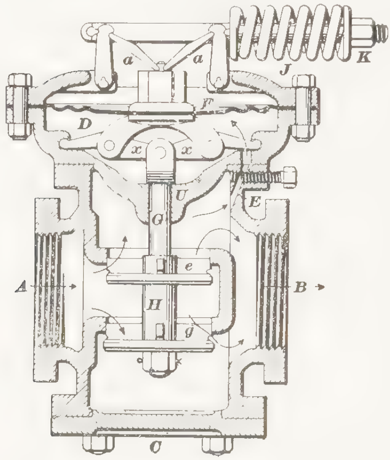


FIG. 21.

the pressure to which the valve will reduce the entering steam may be adjusted by increasing or diminishing the resistance of the spring *J*. To allow this to be readily done, a nut *K* is provided. The device illustrated will not only reduce the pressure, but will also regulate it automatically; that is, although the pressure of the entering steam may vary considerably, as long as its pressure does not fall below the pressure for which the valve is set, it will give a practically uniform pressure on the discharge side. A steam gauge should be fitted to the steam pipe on the discharge side.

THE ECONOMIZER.

72. Economizers make use of the heat in the waste furnace gases to raise the temperature of the feedwater. The temperature of the gases on entering the chamber is usually from 450° to 650° F. By lowering the temperature to 250° or 300°, a marked saving of fuel must result. The draft of the chimney, however, depends on the temperature of the gases. The loss in draft that results from the reduction of the temperature of the flue gases may be made up by increasing the height of the chimney.

73. Fig. 22 shows the position of an economizer with respect to boilers and chimney. As will be seen, it is placed directly in the flue. The water enters at *F*, where the economizer is coolest, and flows along the pipe *G*. From *G* the water flows out at right angles to its former direction through a series of horizontal parallel headers *K* and up the rows of vertical tubes *H*, *H*, etc. that connect with them. Each of these vertical rows has an upper header *N*, *N*, etc. that has an outlet into the delivery pipe *I* to which is connected the pipe *J* leading to the boilers *L*, *L*. The hot gases from the boilers pass through the rows of tubes on their way to the chimney, coming into contact with the rows containing the hottest water first. The feedwater may be heated by this means to as high

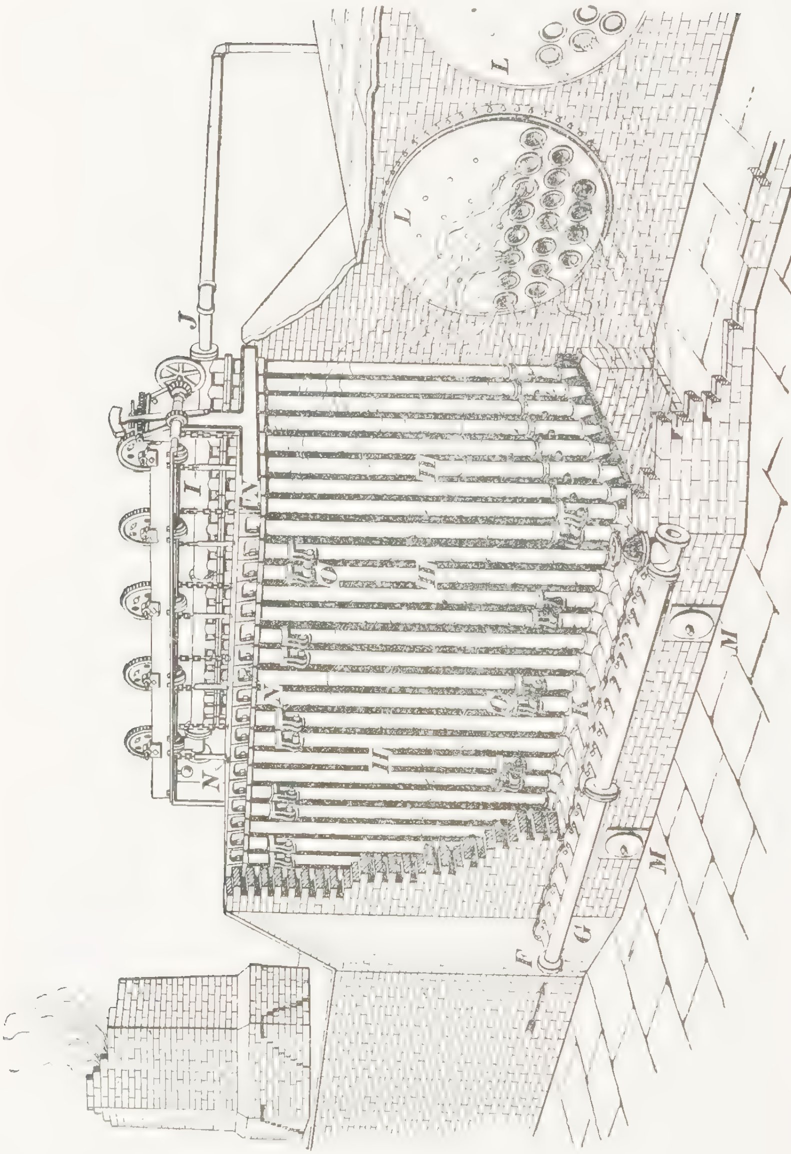


Fig. 22.

as 300° F. and the temperature of the gases reduced from the neighborhood of 600° to 250° or 300°.

74. The hot gases deposit soot and other unconsumed particles upon the tubes. Since soot is a very bad conductor of heat, the efficiency of the economizer will soon be greatly impaired unless means are provided for removing the soot. This is accomplished by scrapers *O*, *O*, etc., which are moved up and down by means of a suitable mechanism on top of the economizer. *M*, *M* are openings for the removal of the soot scraped from the tubes.

The piping should always be arranged so that the cold water may be pumped directly into the boiler, if necessary, without first passing through the heater or economizer. Then, if the heater gets out of order, the boiler may be still used.

THE RECORDING GAUGE.

75. As implied by the name, a steam-pressure recording gauge is a device that produces an accurate and complete record of the steam pressure, giving all the variations with the exact time and duration of each change. These changes are registered on a chart, which is generally changed day by day and filed away for future reference.

76. In all recording gauges there is a pencil attached to a suitable spring-loaded mechanism, similar to a steam-engine indicator; the pencil moves up and down with the variation in pressure and traces a line on a chart placed on a drum or the face of a wheel that is revolved by clockwork.

77. A recording gauge not only settles, beyond the possibility of any dispute, the question of what the steam pressure was at a particular time, and thus allows the blame for any accident or other happening to be traced to the right source, but it also tends to insure greater economy in the operation of the plant, as it exhibits clearly, by the deviation from a straight line of the line traced by the pencil, any neglect on the part of the fireman.

SELECTION OF BOILERS.

INTRODUCTION.

78. The selection of a type of boiler for a prospective plant or for one that already exists depends on several things that should be carefully considered before a decision is made. There is no doubt that in some cases it is difficult to make a selection of a type to meet all the requirements, but then the relative merits of the individual considerations should be weighed rather than the considerations themselves. The principal considerations with which we have to deal when selecting a type of boiler for a given plant are given in the following articles.

79. Nature of the Feedwater.—Waters that abound in scale-forming matter should decide in favor of the plain cylindrical or the horizontal tubular boiler, because of the comparative ease with which they can be cleaned at a minimum cost. Water-tube boilers using such waters rapidly become "scaled"; the scale can only be removed in most cases (without injuring the boiler) by the application of expensive appliances, which, considering the type of boiler, consume considerable time in the operation.

80. Kind of Service.—Boilers that are to be installed in buildings in which there are a number of people should preferably be of the water-tube type, because of the comparative safety of this type. Boilers for sawmills and similar places should preferably be of the simplest construction and of the fire-tube type, because of the fact that they will not receive the care that a water-tube boiler requires. For railway power stations, where the rapid raising of steam is frequently demanded, the water-tube boiler should be selected, because of its superiority in this respect. Boilers that are in almost constant service, where time for repairing, cleaning, and overhauling is extremely limited, should be of the horizontal tubular type, because of its ability to stand such

service for a longer period of time with the minimum amount of overhauling than most other types.

81. Influence of Available Labor.—This is an important consideration and cannot be given too much thought. The water-tube boiler requires more care than the fire-tube boiler, which is so largely used in stationary work. A plant in which there is only one attendant should in general not be equipped with water-tube boilers, because the attendant will not have the requisite amount of time to properly care for them. On the other hand, plants that have an attendant just for the purpose of taking care of boilers could safely be equipped with water-tube boilers, as far as this consideration is concerned. Experience has shown that it requires a larger force of men to operate and maintain the water-tube boiler than it does for any other type. This statement is contrary to the claims of makers of water-tube boilers; however, it represents the opinion of many operating engineers.

82. Available Space.—This consideration alone frequently leaves no choice in the matter. For shallow basements and out-of-the way corners, no boiler is as suitable as the horizontal return tubular. Of course, where space is plentiful, other considerations may cause a different type of boiler to be chosen.

83. Steam Pressure to be Carried.—Water-tube boilers are best adapted for high pressures, because they are stronger. Take, for instance, the tubes; in a water-tube boiler the pressure is internal, while in the fire-tube boiler or flue type, the pressure is external, tending to collapse the tube or flue. Now, since for equal thicknesses and diameters a cylindrical body will collapse under less pressure than that which will tear it asunder, it follows that the water-tube boiler, with tubes equal in size to those of a fire-tube boiler, will stand safely a higher working pressure.

84. First Cost.—When first cost is the principal consideration on which a selection is made, the plain cylindrical

and horizontal tubular boilers are the most economical to purchase. But sometimes the practice of such initial economy proves to be the most expensive in the end, so that this consideration should not be given too much weight apart from the others. From this it must not be inferred that the most expensive boiler (first cost) is always the most economical. Other considerations and conditions bear largely on the question.

85. Expense of Operation and Maintenance.—The principal item affecting the cost of operation of the different types of boilers is the evaporative efficiency of each. The boiler that has the highest efficiency will cost the least for operation, assuming other things to be equal. With regard to repairs, it may be said that in general the water-tube boiler is the most costly, if it is to be kept in first-class condition.

86. Influence of Location.—It would not be wise to instal a boiler whose construction demands frequent overhauling and repairing to keep it in thorough condition in a place remote from where such work could be done by skilled hands and with the proper appliances. Such a location demands the simplest make of boiler and that which is the least liable to require extensive overhauling and repairs.

BOILER MANAGEMENT.

GETTING UP STEAM.

FILLING THE BOILER.

1. Precautions.—Assuming that an old boiler has been cleaned or that an entirely new boiler is all ready to be filled and started, the course of action described below, which is practiced by many engineers, is recommended.

2. Before filling the boiler it should be carefully examined internally, to see that no tools, lamps, or other foreign matter has been allowed to remain. In a new boiler particularly, care must be taken that no oil is left on any of the internal surfaces in contact with water, wiping the surfaces clean. The presence of oil is due to the leakage of lamps or cans that the workmen have been using during construction. Should oil be allowed to remain in the boiler, it would tend to prevent the water from coming in contact with the metal, which condition is liable to lead to an overheating of the parts exposed to the fire. In connection with this it may be mentioned that with an increase of temperature above 600° F., iron or steel becomes weaker; hence, it is important to constantly keep the water in contact with one side in order to prevent a dangerous increase of temperature of the metal.

3. Closing the Boiler.—Before starting the water into the boiler, any of the manhole plates or handhole plates that

have been removed preparatory to cleaning and overhauling must be replaced. It must be observed that the gaskets (or **joints,** as they are sometimes called) are in good condition and also the surfaces with which such gaskets come in contact. It is usual to place a mixture of cylinder oil and graphite (black lead) on the outer surface of the gasket, so that it may be removed without tearing; the plate and gasket then come off together as one piece, thus allowing the gasket to be used again. It is important that the manhole plates and handhole plates be properly replaced and secured in order to prevent leakage. Such leakage can rarely be stopped by tightening and will generally necessitate blowing the boiler down and a remaking of the joint. It is advisable to avoid such mishap with its attendant delay by making the joints carefully, using new gaskets if there is the least doubt about the old ones. The blow-off valve should be closed in case inspection shows it to be open.

4. Amount of Water Required.—The boiler should be filled until the water shows half way up in the gauge glass, assuming that the latter has been correctly placed. The attendant should satisfy himself of this matter when first taking charge of a boiler or boilers. In any case, the water should be run up high enough to cover all parts of the boiler that are subjected to the action of the fire and hot gases on the outside.

5. Methods of Filling the Boiler.—In some cases the water can flow in and fill the boiler to the required height by the pressure which exists in the main supply pipe, the piping being arranged to permit this. In other cases it may be necessary to use a hose or to fill with a steam pump or hand pump, the location of the plant and existing conditions determining just which method is to be practised.

6. Escape of Air.—While filling a boiler it is necessary to make provision for the escape of the contained air, since otherwise the pressure caused by the compression of the air may prevent the boiler from being filled to the proper

height. Most boilers have some valve that can be used for this purpose; generally the safety valve may be raised or the top gauge-cock may be left open until steam issues therefrom, when it may be closed. Sometimes the manhole plate is left off while filling a boiler; the air then escapes through that opening.

MANAGEMENT OF FIRES IN STARTING.

7. Precautions.—After the boiler has been filled and before starting the fire, the attendant should see that the water column and connections are perfectly clear and free, i. e., that the valves in the connections and the gauge-glass valves are open so that the water level may show in the glass; he should also see that the gauge-cocks are in good working order and open the top cock or the safety valve; he should take care that the stress on the stop-valve spindle is relieved by just unscrewing the valve from the seat without actually opening it. If this were not done, the unequal expansion of the parts during the raising of steam would tend to unduly strain some of the weaker parts, and, furthermore, the valve would be difficult to open. He should make sure that the pump, or injector, or whatever device is used to feed the boiler, is in good working order, and ready to start when required. In general, he should see that everything is in readiness.

8. Laying and Starting Fires.—It is usual to cover the grates with a layer of coal (if coal is the fuel used) first, and then add the wood, among which may be thrown oily waste or other combustible material that may be at hand. To start the fire, light the waste or other easily ignited material and open the damper and ash-pit doors to produce draft. Then close the furnace door. After the wood has been well started to burn, spread it evenly over the grate and add a fine sprinkling of coal, until this in turn begins to glow, when more coal may be added and the fire occasionally leveled until the proper thickness of fire has been obtained. It sometimes happens that the chimney refuses to draw;

the draft can generally be started, however, by building a small fire in the base of the chimney.

9. Importance of Slow Fires.—It should be borne in mind that when getting up steam the fire should not be forced, but, instead, should be allowed to burn up gradually, thus giving the boiler a chance to expand more uniformly under the action of the increasing heat. By forcing the fire, the plates or tubes that are nearest the fire suffer extreme expansion, while those parts that are remote from the fire are still cold; under such conditions the seams and rivets, and also the tube ends, which are expanded into the tube plates, are liable to be severely strained, and, possibly, permanently injured. It is not desirable to raise steam in any boiler (excepting in steam fire-engines, which are so constructed as not to be materially injured by so doing) in less than from 2 to 4 hours, according to the size, from the time the fire is first started. When steam begins to appear from the top gauge-cock or raised safety valve, as the case may be, then it may be closed and the pressure allowed to rise, but still slowly, until the desired pressure has been reached.

10. Trying the Attachments Under Steam.—After the pressure at which the boiler is to run has been reached, before cutting it into service, all the valves and cocks should be tried under pressure. The safety valve should be raised and its action noted; the water column should be blown out and the gauge-cocks tested; the feeding apparatus should be tried; and it should be noted particularly if the check-valves seat properly and the valve in the feedpipe is open. All the accessible parts should be examined for leaks. Everything proving to be in good order, the boiler is now ready to cut into service.

CONNECTING BOILERS.

11. Cutting a Boiler Into Service.—Cutting a boiler into service is accomplished by opening the stop-valve, thus permitting the steam to flow to the engine or other mechanism. The stop-valve, and in fact any valve that is

subjected to great pressures, should be opened very slowly to prevent too sudden a change in the temperature and expansion of the piping through which the steam flows and to prevent *water hammer*. The latter is caused by large bodies of condensed steam being thrown violently forwards by the inrushing steam, due to opening a valve too quickly. Water hammer is liable to prove disastrous to the piping, the heavy blow due to the momentum of the body of condensed steam moving with a high velocity being likely to cause a leaking of the joints, if not a breaking of the pipes. To prevent the accumulation of water, the steam-pipe drain should be kept open until the pipe is thoroughly warmed up; that is, until nothing but steam issues from the drain. In large plants with many boilers and long steam mains it takes several hours to thoroughly warm these pipes by a slow circulation of the steam, and not until then is the main stop-valve fully opened.

12. Connecting Boilers of a Battery.—Before connecting the different boilers of a battery together, i. e., to the same steam main, the precaution of equalizing the pressures in the different boilers must be observed in order to prevent a sudden rush of steam from one boiler to another. The pressures should all be equal within a variation of, say, 2 pounds before an attempt is made to connect the boilers.

13. Changing Over.—In plants where there are duplicate sets of boilers, one set being in operation while the other is undergoing repairs, overhauling, and cleaning, the method of changing over, or connecting, is as follows: Start fires and raise steam in the boilers that are to be cut into service. Allow the pressure to rise in all to within 5 pounds of that which is in the boilers in operation; for example, suppose the gauge pressure carried in the boilers in operation be 100 pounds, then allow the rising pressure to reach 95 pounds. The boilers will then be all ready to change over, or connect.

14. All arrangements before changing over should be made with a view to getting all the heat that can be obtained

from the fires in the boilers that are to be cut out. This can be accomplished by running until the fires have given up all of their available heat for making steam, as indicated by the gradual fall in pressure when the dampers are wide open, and then making the change. While the fires in one set of boilers are burning low and the pressure is falling, the pressure in the boilers to be cut in is gradually rising and meeting, so to speak, the falling pressure of the set in operation. When the difference of 5 pounds is reached, change over. A man should be stationed at each stop-valve, and while one is being opened the other should be closed; the engine will continue running uninterruptedly while the change is being made.

15. The boilers having been cut in, it will be in order to change the appliances over, such as the engine-room steam gauge, which is in connection with both sets of boilers, valves being the medium through which the connection is made, and the feed apparatus, etc. To do this, the valves must be closed on the boilers that have been cut out and opened on the set that has been cut in, thus connecting the gauge, etc. to the boilers in operation.

16. The reason a difference in pressure is suggested at the time of changing over is because of the fact that the water of condensation at the dead end of the steam pipe will flow into the boilers to be cut in when the stop-valves are opened, instead of rushing *towards the engine*, as it would otherwise do. This water running into the boilers can do no harm, while the same cannot be said were the conditions reversed.

17. After a set of boilers has been cut out and all the valves that require it are closed, it should be allowed to stand at least one day before commencing overhauling. Should the boiler be immediately emptied (blown down) and opened up after being cut out of service, the rapid contraction of the parts cannot be other than detrimental to the life of the boiler. Furthermore, if the boiler with its

contained water is allowed to cool down gradually, the sediment will not be baked hard. Of course, it will be understood that the foregoing instructions are general in character and they may have to be modified to meet existing conditions in certain plants, when the good judgment of the operator will be in demand.

MANAGEMENT WHILE RUNNING.

EQUALIZING THE FEED.

18. When the boilers of a battery have been cut into service and hence are all connected together through the steam main, the regulation and equalization of the feedwater becomes an important factor in the general management while the boilers are in operation. Each boiler has its own check-valve and feed stop-valve, and generally all the boilers are supplied from one pump, which is running constantly. The quantity of water admitted to each boiler is regulated by its feed stop-valve. When the water gets low in any boiler, as indicated by the gauge glass, its feed stop-valve is opened wider, thus permitting a greater quantity to enter in a given time, while at the same time the feed stop-valves on one or more of the other boilers in operation may be closed partially and thus divert the feedwater to the one most requiring it. Some boiler plants have check-valves with an adjustable lift; in that case the feed is equalized generally by adjusting the lifts of the check-valves, the stop-valves being left wide open while running. It will be understood from the foregoing that the object in view is the maintaining of an equal water level in all the boilers through the manipulation of the feed stop-valves or check-valves. A boiler that is not doing its legitimate share of the steam producing may be known by the fact that the feed stop-valve or check-valve on that boiler will be nearly if not quite closed most of the time.

THE FIRES.

19. General Considerations.—The safe and economical operation of steam boilers calls for careful and intelligent management. The fires should be kept in such condition as to maintain the desired pressure and to burn the fuel with economy. Different fuels require different handling and hence only general rules can be given; much will depend on the skill and judgment of the attendant, who must himself discover in each case by actual trial the best method to adopt and pursue. The fires require to be cleaned at intervals; the time and method of so doing depend on conditions, such as the nature of the fuel, the rapidity with which it is being consumed, the style of grate in use, and the construction of the furnace. Here much is left to the choice and judgment of the attendant, who should readily discover what is best to be done in any particular case.

20. Cleaning Fires.—There are two methods employed in cleaning the fires: First, that of cleaning the front half and then the rear half; second, that of cleaning one side and then the other side of the fire.

21. In the first method, previous to cleaning, green fuel is thrown on and allowed to partially burn until it glows over the entire surface. The new and glowing fuel is then pushed to the back of the furnace with a hoe, leaving nothing on the front half of the grate but the ashes and clinkers, which are then pulled out, leaving the front end of the grate entirely bare. The new fire which had been pushed back is now drawn forwards and spread over the bare half of the grate. The ashes and clinkers that are on the rear half of the grate are then pulled over the top of the front half of the fire and out through the furnace door; this leaves the rear half of the grate bare, which must be covered by pushing back some of the new front fire. The clean fire having been spread evenly, some new fuel must be spread over the entire surface.

22. The second method referred to is substantially the same in principle as that just described, with this difference,

the fire is pushed to one *side* instead of to one *end* of the furnace, as in the first method described.

23. The condition of the fires themselves and the nature of the service of the plant will soon determine just how often and at what time the cleaning of fires should take place. In general, the fires in stationary boilers require cleaning at intervals of from 8 to 12 hours. Fires require cleaning more often when forced draft is used than when working with natural draft.

24. Importance of Rapid Cleaning.—Rapidity in cleaning is of great importance, since during cleaning a large volume of cold air enters the furnace and chills the metallic surfaces with which it comes in contact; consequently, the boiler suffers, even though it be in a small degree. It is the greatest advantage of shaking grates that they allow the fire to be cleaned without opening the furnace door; the inrush of cold air and consequent chilling of the plates, etc. is thus avoided.

25. Preparing for Cleaning.—The steam pressure and the water level should be run up as high as is safe and the feed should be shut off before starting to clean fires, in order to reduce the loss in pressure while cleaning. The condition of the fire during cleaning and the open furnace doors cause the pressure to drop quite rapidly, but the rapidity and the amount of drop will be reduced by taking the precautions mentioned and cleaning rapidly.

26. Drop in Pressure While Cleaning.—The amount of drop in pressure while cleaning fires depends on several conditions that may exist and the effects of which should be understood by the attendant in order that he may govern himself accordingly. Every plant has its own individuality in this respect, which depends on the local conditions, and these must be studied. For example, with a boiler that has a small steam space and in addition is too small for the work required of it without forcing, it is to be expected that the

drop in pressure will be much more than if the reverse conditions exist. Furthermore, it may be necessary to clean fires while steam is being drawn from the boiler, instead of being able to clean at a time when the engine is stopped. In that case a greater drop must be expected than when cleaning while no steam is drawn from the boiler. Whenever it is possible, it is advisable to do the cleaning at a time when no steam is being drawn from the boiler or when the demand for steam is light.

27. When cleaning fires in a battery of boilers, it should be the aim to clean them in rotation, in order that the drop in pressure may be small.

CARRYING A UNIFORM PRESSURE.

28. Benefit Derived.—The attendant should aim to carry the pressure in his boiler as uniform as possible and not allow it to fall below the standard at one time and then permit it to rise above it at another time. By the "standard" is meant that pressure which has been predetermined on to operate the plant. If 100 pounds gauge pressure is the standard desired or required, he should try to sustain that pressure throughout the day.

29. The reason why a steady steam pressure and a steady water level are conducive to economy in the use of fuel is to be found in the fact that with these conditions in a properly designed plant there will be a fairly steady temperature in the furnace, which, under normal conditions, is sufficiently high to insure a thorough ignition of the volatile matter in the coal. Now, with a constant demand for steam, a fluctuation in the steam pressure is caused by a change in the furnace temperature (assuming the feedwater supply to be constant), and whenever the steam pressure is down, the furnace temperature is low at the same time. In consequence of this, large quantities of the volatile matter in the coal often escape unconsumed and a serious loss of heat is thus caused. Furthermore, with a steady steam pressure the stresses on the boiler are constant, and since

under a constant stress the deterioration of iron and steel is much less rapid than under varying stresses averaging the same as the constant stress, the deterioration will be reduced, and hence the life of the boiler will be increased and repair bills will be smaller than otherwise.

30. Directions.—During the period of time between the cleaning of the fires, the pressure may be carried nearly uniform by observing the following instructions: Manipulate the feeding apparatus so that just the necessary amount of water constantly enters the boiler, and thus maintain a constant level. If intermittent feeding is practiced (because of local conditions, as, for example, on account of an injector or pump that is so large that it would be impossible to run them continuously without materially increasing the height of the water level), the time to stop feeding is just before firing; that is, do not feed while firing nor resume feeding until the new fire begins to make steam, as indicated by the rise of pressure on the gauge.

31. If the pressure tends to rise above the standard, partially close the dampers and increase the quantity of feed, assuming here that no damper regulator is fitted and that hence the damper is regulated by hand. A damper regulator, proper firing, and constant feeding are essential for carrying a practically uniform pressure. Should the pressure continue to rise, throw on more green fuel, close the damper, increase the feed, and only as a *last resort* open the furnace door.

32. A uniform steam pressure cannot be kept without proper firing. The following directions should be observed: Keep the fire of uniform thickness and allow no air holes in the bed of fuel. Fire evenly and regularly and not too much at a time. Keep the fire free from ashes and clinkers and as clean at the corners and sides as at the center. Keep the ash-pit clear. Do not clean the fires oftener than necessary.

33. In connection with maintaining a constant water level, observe the following: The first duty of the fireman

on going to work should be to examine the water level. The gauge-cocks should be tried; the gauge glass is not always reliable. In a battery of boilers the gauge-cocks on *each* boiler should be tried. Some serious explosions have resulted from the fact that the fireman only consulted the water level in the first boiler and took it for granted that the water level in the other boilers was the same.

34. Should the pressure get below the standard at times other than while cleaning the fire, open the damper and shut off the feed, and in plants where there are many pumps and auxiliary engines, shut off one or more of those that are not absolutely required to be constantly running.

35. The person in charge of the boiler must modify the foregoing instructions to suit the local conditions of the plant in which he is.

PRIMING AND FOAMING.

36. Priming.—Priming is analogous to boiling over; the water is carried into the steam pipes and thence to the engine, where considerable damage is liable to take place if it is not checked in time. There are several causes for priming, of which the most common ones are the following:

1. Insufficient boiler power;
2. Defective design of boiler;
3. Water level carried too high;
4. Irregular firing;
5. Sudden opening of stop-valves.

37. When the boiler power is insufficient, the boiler is forced in order to furnish enough steam, and, consequently, the steam bubbles rise through the water at such speed that they carry particles of water with them by friction and adhesion. The best remedy is to increase the boiler plant; the next best thing to do is to put in a separator, which obviously will only prevent the entrained water from reaching the engine, but will not stop the priming.

38. Defective design of a boiler generally consists of too small a steam space or a bad arrangement of the tubes, which may be spaced so close in an effort to obtain greater heating surface as to interfere seriously with the circulation. In horizontal return-tubular boilers, a small steam space can be cured by the addition of a steam drum; sometimes the top row of tubes can be taken out to advantage, which permits a lower water level. Defective circulation is difficult to detect and to remedy; if believed to be due to too close a spacing of the tubes, a marked betterment has occasionally been effected by the removal of one or two vertical rows of tubes.

39. The remedy for too high a water level is obvious—carry the water at a lower level. With irregular firing, especially when the draft is strong, the evaporation rate will be so high at times that the steam bubbles will rise at such speed as to carry water with them, just as in the case of insufficient boiler power. The sudden opening of a stop-valve means a momentary local lowering of the pressure near the steam outlet from the boiler, in consequence of which some of the water in the other parts of the boiler will, by the greater pressure, be thrown towards the outlet and mix with the steam rushing from the boiler.

40. Priming manifests itself first by a peculiar clicking sound in the cylinder of the engine, due to the water being thrown against the heads. In cases of very violent priming, the water will suddenly raise several inches in the gauge giass, thus showing more water in the boiler than there really is. When priming takes place, it can be checked temporarily as follows: Close the damper, and thereby check the fires until the water is quiet; the engine stop-valve should also be partially closed to check the onrush of water. Observe if the water drops in the gauge glass, and then if more feed is needed, increase the feed. To prevent damage to the engine, open the cylinder drains. Regular and even firing tend to prevent priming.

41. Foaming.—Foaming is not synonomous with priming, though frequently considered so. Foaming is the result of dirty or greasy water in the boiler, and, as the name implies, the water foams and froths at the surface, but does not lift, as in the case of priming. A boiler may prime and foam simultaneously, but a foaming boiler does not always prime. Foaming while taking place is visible in the glass gauge and is best cured by using the surface blow-off. If no surface blow-off is fitted, the bottom blow-off may be used in order to get rid of the dirty water. Foaming, the same as priming, will cause a wrong water level to be shown, and hence the first thing to do in case of foaming is to quiet the water by checking the outrush of steam, either by slowing the engine down or checking the fire, or by both.

SHUTTING DOWN FOR THE NIGHT.

42. Preparations.—Before shutting down for the night it is advisable to fill the boiler to the top of the glass, so as to be sure to have sufficient water to start up with in the morning. The presence of possible leaks through the valves, tube ends, or seams necessitates this course of action being taken. Even if no leaks exist, it is a good practice to do this, if for no other reason than to admit of blowing out a portion before raising steam in the morning. All the gauge-cocks should be tried and the water column should be blown out to insure their being free and clear.

43. Banking Fires.—The fires may be banked at such a time that there will be about enough steam to finish the day's run, thus shutting down under a reduced pressure with only a remote possibility of it rising again through the night. If the fires are properly banked and the steam worked off while the feed is on, the possibility of the pressure rising during the night to a dangerous extent will cease to exist.

To bank the fires, they should be shoved to the back of the grate and well covered with green fuel, leaving the

front part of the grate bare, thus preventing any possibility of the banked fire burning up through the night.

44. Valves to be Closed.—The steam stop-valve, feed stop-valve, whistle valve, and other steam valves should be closed; the valves at the top and bottom of the gauge glass also should be shut off to prevent loss of water, etc. in case the glass should break through the night.

45. Final Survey.—A careful attendant will take a walk around the entire plant and see that all the valves and cocks and other attachments of the boiler are all right before leaving it. Even though the feed check-valves be in good condition, it is still desirable to close the feed stop-valves as a matter of precaution.

46. Manipulation of Damper.—If there is a damper regulator, it should be so arranged that the damper may be left *closed*, but not quite tight, because a small opening must be left to permit the collecting gases from the banked fire to escape up the chimney, otherwise there is danger of the accumulated gas igniting and causing an explosion sufficient in force to at least wreck the boiler setting. It is very important that this precaution be taken and a certain mark be made by which the distance the damper is open can be seen at a glance. In fact, a damper should be so made that when shut to the full extent of its travel, there will be still sufficient space around it to allow the gas to escape. The damper regulator should be rendered inoperative in any manner permitted by its design.

47. Modifications of Directions.—In the foregoing instructions with regard to shutting down for the night it has been assumed that no person is in the plant during the night, except possibly a watchman, who may not understand anything at all about a boiler. Should a plant be left in charge of a watchman capable of acting as a night fireman, the foregoing directions may be modified accordingly. For instance, in such a case it would be preferable to leave the gauge-glass valves open, so that the water may be seen at

any time. Should the glass break, no serious quantity of water will be lost, because a new one can be placed in by the man in charge.

STARTING IN THE MORNING.

48. Starting Up the Fires.—When entering the boiler room in the morning, the first thing to give attention is the quantity of water in the boiler. The gauge glass and the gauge-cocks should be tried and the water level noted. After it has been found that the water is not too low, the banked fires may be pulled down and spread over the grates and allowed to come up slowly, the damper regulator, if one is fitted, in the meantime having been connected.

49. Blowing Down.—While the fires are burning and before the pressure begins to rise, the blow-off cocks (or valves) should be opened and the boiler blown down some; that is, a small quantity of the water should be blown out, say 3 or 4 inches, as shown in the glass gauge. This should be done every morning, so that any impurities in mechanical suspension in the water that settled during the night may be blown out. Great care should be exercised while blowing down that too much water is not blown out. Under no circumstances must the attendant leave the blow-off while it is open. Disaster to the boiler is liable to follow a disregard of this injunction. Next, all the valves, except the stop-valve, which were shut the night before should be opened and tried to see that they are free and in good working order.

50. Turning on the Steam.—After the feeding apparatus has been tried and found all right and everything is ready for a start, the stop-valve may be opened for the day's run. It should be remembered that all steam valves about the boiler should be opened *slowly*, and it is desirable that they be closed slowly also, especially the large main valves.

INSTRUCTIONS FOR BOILER ATTENDANTS.

51. The following instructions issued by the Hartford Steam Boiler Inspection and Insurance Company will serve to guide the fireman or attendant in the management of boilers, said instructions being general in character:

1. *Firing.*—Keep the fire of uniform thickness and allow no air holes in the bed of fuel. Fire evenly and regularly, and not too much at a time. Keep the fire free from ashes and clinkers and as clean at the corners and edges as at the center. Keep the ash-pit clear. Do not clean the fires oftener than necessary.

2. *Water Level.*—The first duty of the fireman on going to work should be to examine the water level. The gauge-cocks should be tried; the gauge glass is not always reliable. In a battery of boilers, the gauge-cock on each boiler should be tried. Some serious explosions have resulted from the fact that the fireman only consulted the water level in the first boiler and took it for granted that the level in the other boilers was the same.

3. *Low Water.*—If the water is discovered to be low, quickly cover the fire with ashes, or, if they are not convenient, with fresh coal. Do not turn on the feed and do not tamper with the safety valve or any other steam outlet. The fire may be drawn as soon as it can be done without increasing the heat.

4. *Foaming or Priming.*—In case of foaming, close the throttle valve of the engine or the stop-valve of the supply pipe, and keep it closed long enough to show the true water level. The foaming can be generally stopped by blowing off and feeding fresh water. In case of violent foaming, due to dirty water, check the draft and cover the fire with fresh coal.

5. *Leaks.*—When leaks are discovered, they should be repaired at once.

6. *Blowing Off.*—When blowing off, the steam pressure should not be over 20 pounds. The boiler should be emptied at least every 2 weeks and filled up afresh. Once every week

would be better. If the water is muddy, blow out 6 or 8 inches every day. A surface blow-out should be opened for a short time at frequent intervals. Examine the blow-out cock and the check-valve every time the boiler is filled; a leakage from either may lead to serious results.

7. *Filling Up.*—Allow the boiler to become cold before pumping in cold water; the practice of filling a hot boiler with cold water causes leaks and fractures and sometimes explosions.

8. *Safety Valves.*—Raise the safety valves from their seats cautiously and frequently. Do not allow the valve to be overloaded.

9. *Pressure Gauge.*—The pressure gauge should stand at 0 when the steam pressure is off, and it should indicate the blowing-off pressure when the safety valve is in action. If the gauge does not do this, it should be compared with the standard gauge, and if wrong, corrected.

10. *Gauge-cocks and gauge glasses* should be kept clean and should be in constant use. The water gauge should be blown out frequently and the glasses and passages to gauges kept clean. An obstructed gauge sometimes shows a false water level. The Manchester Boiler Association attributes more accidents to this cause than all others combined.

11. *Feed-Pumps and Injector.*—Both pump and injector should be of ample size, and whichever is used should be made to work as uniformly and continuously as possible. It is better to have two independent means of feeding the boiler. Check-valves should be frequently examined.

12. *Removal of Sediment and Incrustation.*—Scale and sediment should be frequently removed. In tubular boilers particularly, the handhole or manhole should be frequently opened and the sediment removed from the portion of the plate over the furnace. Care should be taken to keep the boiler as free as possible from incrustation.

13. *Cleaning.*—All heating surfaces should be kept free from soot and dirt. Tubes should be cleaned often.

14. *Exterior of Boiler.*—Care should be taken that no water comes in contact with the exterior of the boiler, either from leaky joints or other sources. Avoid dampness in the setting or in the covering of the boiler. Dampness leads to external corrosion.

15. *Blisters and Cracks.*—A blister should be examined at once and trimmed or patched. If a plate is badly cracked, it must be renewed.

16. *Fusible Plugs.*—Fusible plugs should be examined when the boiler is cleaned and scraped clean on both sides; otherwise they are liable to prove worthless.

17. *Air Leaks.*—See that the furnace, combustion chamber, and smoke flue are tight. The admission of air through the brickwork of the setting is sometimes the source of considerable loss.

18. *Galvanic Action.*—Examine the parts of the boiler where brass or copper and iron come in contact in the presence of water. Galvanic action may produce corrosion under such circumstances; if such be the case, the corrosion may be prevented by placing pieces of zinc in the boiler.

19. *Rapid Firing.*—Steam should be raised slowly in boilers with thick seams or seams exposed to the fire. Otherwise overheating may occur.

20. *Cleanliness.*—The boiler room, boiler, and mountings should be kept clean and in good order.

21. *Unused boilers* may be kept in good condition by filling them full of water in which a quantity of common washing soda has been placed. Another method is to empty the boiler, dry it thoroughly, place trays of quicklime in the bottom, and seal as nearly air-tight as possible. The latter method is often used for marine boilers.

52. All these instructions are excellent; the only item that any exception can be taken to is item 6. It is much better to allow the boiler to cool down entirely than to blow down under steam pressure, in order to prevent the heat from baking hard any sediment contained in the boiler.

WEAR AND TEAR.

CORROSION.

INTERNAL CORROSION.

53. **Corrosion** may be defined as the eating away or wasting of the plates due to the chemical action of impure water. It is probably the most destructive of the various forces that tend to shorten the life of the boiler. Corrosion is of two forms—*internal* and *external*. **Internal corrosion** may present itself as: (1) *uniform corrosion*, (2) *pitting* or *honeycombing*, (3) *grooving*.

54. In cases of **uniform corrosion,** large areas of plate are attacked and eaten away. There is no sharp line of division between the corroded part and the sound plate, and oftentimes the only way of detecting the corrosion is to drill a hole through the suspected plate and thus ascertain its thickness. Corrosion often violently attacks the staybolts and rivet heads.

55. **Pitting** or **honeycombing** are readily perceived. The plates are in spots indented with holes and cavities from $\frac{1}{32}$ to $\frac{1}{4}$ inch deep. The appearance of a pitted plate is shown in Fig. 1. Upon the first appearance of pitting, the surface so affected should be thoroughly cleaned and a good coating of thick paint made

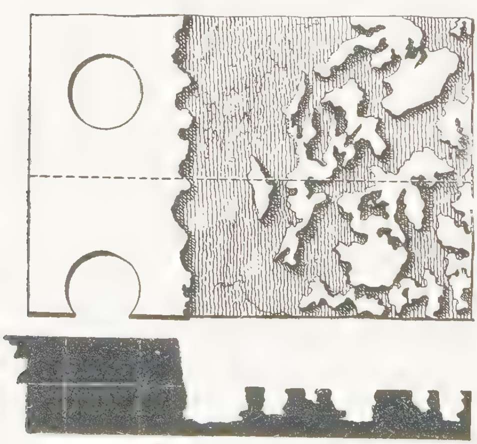


FIG. 1.

of red lead and boiled linseed oil be applied. This treatment should be given from time to time to insure protection to the metal.

56. Grooving is generally caused by the buckling action of the plates when under pressure. Thus, the ordinary lap joint of a boiler distorts the shell slightly from a truly cylindrical form, and the steam pressure tends to bend the plates at the joint. This bending action is liable to start a small crack along the lap, which, being acted on by corrosive agents in the water, soon deepens into a groove, as shown in Fig. 2. The mark made along the seam by a sharp calking tool, when used by careless workmen, is almost certain to lead to grooving.

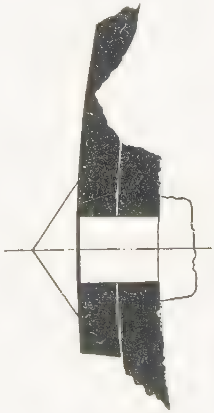


FIG. 2.

57. To prevent corrosion, the feedwater should be as free as possible from corrosive impurities. When bad water must be used, the corrosive impurities must be neutralized by adding alkaline substances, such as caustic soda or soda ash.

58. Zinc is used to prevent corrosion in marine boilers. It is believed that corrosion is due in some measure to galvanic action between the non-homogeneous portions of the iron or steel plates. By placing the plates in connection with slabs of zinc, a galvanic action is set up between the iron and the zinc, which destroys the latter and leaves the former untouched. In the British Navy zinc slabs 12 inches by 6 inches and $\frac{1}{2}$ inch thick are attached to the boiler stays, there being one slab to every 20 horsepower. These are eaten up and renewed about every 60 to 90 days. The zinc is reported to perform its duty very effectively.

EXTERNAL CORROSION.

59. External Corrosion of Fire-Tube Boilers.—External corrosion frequently attacks stationary boilers, particularly those set in brickwork. The causes of external corrosion are dampness, exposure to weather, leakage from joints, moisture arising from the waste pipes or blow-off, etc. When leakage occurs in a joint that is hidden by the brickwork setting, the plates may be corroded very seriously without being discovered.

60. External corrosion should be prevented by keeping the boiler shell free from moisture and by repairing all leaks as soon as they appear. Joints and seams should be in a position where they may be inspected for leaks.

61. Leakage at the seams may be caused by delivering the cold feedwater on to the hot plates; another cause is the practice of emptying the boiler when hot and then filling it with cold water. The leakage in both cases may be traced to the sudden contraction of the plates due to the sudden cooling. In any case abrupt changes in the temperature of the shell should be avoided. The rush of cold air into the furnace of an externally fired boiler when the door is opened is a fruitful source of leakage and fracture. For this reason the shell should be constructed, if possible, so that none of the seams are in contact with the fire.

62. External Corrosion of Water-Tube Boilers.—In horizontal water-tube boilers of the inclined-tube type, external corrosion principally attacks the ends of the tubes close up to the headers into which they are expanded, especially at the back ends. This is caused by the combined action of leakage and the gases of combustion, which rapidly destroys the tube. This corrosion usually extends to from 4 to 8 inches from the header, and small pin-point holes appear shortly, manifesting

themselves by threadlike streams of water while under pressure.

63. In the course of time the tubes will leak around the expanded portion in the headers, and unless the leak is a large one its presence may not be even suspected. In such type of boilers a small leak around the tubes is hard to locate, unless it is in a tube near the top or bottom rows. Hence, such leaks continue for a considerable time, partly obscured by the accumulation of soot, until the tube becomes eaten away as described before.

64. When upon examination it is found that a leak exists somewhere in a tube near the center row, but which cannot be exactly located, it would be advisable to expand *all* the tubes in that immediate section so as to make sure of embracing the right one. This of course means more labor, but when in doubt as to the exact one, it will pay to do as has been suggested.

65. If leaks are prevented or attended to immediately, no corrosion will take place, as the gases of combustion are harmless, unless acting in conjunction with water or dampness, or unless the coal is rich in sulphur. Should, however, the ends of several tubes be found badly corroded but not yet leaking from that cause, or even if leaking as described, the tube may be saved by introducing a split sleeve or ferrule made as shown in Fig. 3, which also shows its method of application. The sleeve may be made of a piece of old boiler tube and may be given a length of from $1\frac{1}{2}$ to 2 diameters. The piece is split lengthwise in two pieces, and these are bent slightly to conform to the inside of the leaky tube. The piece should be split at an inclination of about $\frac{3}{8}$ inch per foot and the edges be filed. After smearing the inside of the leaky tube with red-lead putty, the one half of the sleeve is put in flush with the tube end;

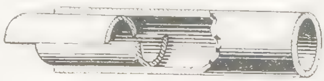


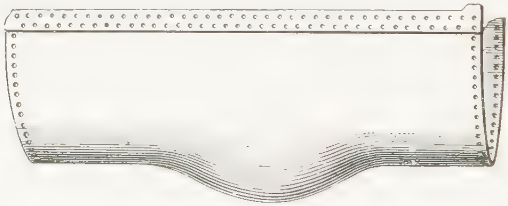
FIG. 3.

the other half is now placed on top, as shown in Fig. 3, and then driven home flush with the tube. By reason of the edges being in a plane inclined to the axis, the two halves of the sleeve are firmly pressed against the inside of the leaky tube when the second half is driven home.

66. The sleeves require to be neatly fitted; in that condition they have been used with good success, effectually stopping a leak, saving the necessity of withdrawing the tube, and also saving the expense of a new one. Obviously they can also be used for fire-tube boilers to stop local leaks.

OVERHEATING.

67. Overheating may be caused by low water or by incrustation. When the plate is covered by a heavy scale, the heat is not carried away by the water fast enough to



A

FIG. 4

prevent a rise of temperature, the plate becomes red hot and soft, and yields to the steam pressure, forming a pocket *A*, as shown in Fig. 4.

68. If the pocket is not discovered and repaired, it stretches until finally the material becomes too thin to withstand the steam pressure; the pocket bursts and an explosion follows. The vegetable or animal oils carried into the boiler from a surface condenser are particularly liable to cause the formation of pockets.

MAINTENANCE OF BOILERS.

OVERHAULING.

TEMPORARY OVERHAULING.

69. Time of Service.—When a boiler has been in continuous service for from 4 to 8 weeks, according to circumstances, it should be cut out of service, thoroughly cleaned both internally and externally, and any necessary repairs made.

70. The period of time that a boiler may be continued in service depends on several things, such as whether the boiler in question is the only one in the plant, the nature of water used, whether or not the plant shuts down Sundays and holidays, and the general local conditions governing the plant.

71. Overhauling on Sunday.—In case there is only one boiler, which is in operation practically all the year, with the exception of Sundays, when it generally rests under banked fires, the cleaning will have to be done at that time. The fires should be drawn out Saturday night and the boiler be allowed to cool down until Sunday morning, when it should be emptied and the manhole plates and handhole plates taken out; it should then be thoroughly washed out by using a hose. In regard to emptying the boiler when not under steam pressure, it may be remarked that the steam space of the boiler must be put in communication with the atmosphere by opening the safety valve, since otherwise the water cannot run out of the boiler. Deposits of mud or other loose substances should be withdrawn with a hoe in preference to washing it out through the blow-off valve, which latter practice is liable to choke the valve and pipe. In the water-tube type of boiler, the mud collects in the mud drum; the mud-drum doors should be removed and the deposit withdrawn through them.

72. Cleaning and repairing boilers during Sunday is at the best not entirely satisfactory, the time being too short. It will do for a while, but there must come a time when such boiler will have to be shut down for a week or so, in order that it can be properly cleaned and repaired. When the time is limited to Sundays only, the attendant should make his plans so that the greatest amount of work can be done in that time.

THOROUGH OVERHAULING.

73. Introduction.—In plants where there are more than one boiler and where it is possible to cut one out of service for a long time, the following method of procedure has been found effective and thorough in its results. The same method of procedure, modified to suit conditions, also applies to boilers out of service but one day a week.

74. Cooling Off.—After the boiler in question has been cut out of service by closing the stop-valve, thus severing its connection with other boilers and the main steam pipe that they supply, and closing the other necessary valves, it is allowed to stand just as left; the spent fires are allowed to die out gradually, the damper being shut, and the water in the boiler to cool off naturally. The boiler should stand thus for at least 24 hours before an attempt is made to commence cleaning operations and repairs.

75. Grates and Ash-Pits.—The first thing to do when the boiler has cooled sufficiently is to pull the ashes from off the grates and out from the ash-pits. The grate bars should be examined and any that may be defective in any respect should be removed, replacing them with new bars. It should be noted if the bearing bars—those that support the entire grate—are secure and in good condition.

76. Furnace Brickwork.—The furnace brickwork—that is, the bridge wall and linings—should next be examined. Where bricks have been knocked out or where walls bulge or are cracked, the extent of such wear and tear should be noted, and when the combustion chamber and

the back connections are cleaned out, repairs should be commenced on the brickwork.

77. Unless the repairs to the furnace brickwork are extensive, the attendant can make them himself; otherwise a regular mason who is accustomed to such work should be called in. In replacing a brick or a number of bricks, care is to be taken that no projecting edges or corners are left, but that each brick is placed flush. The bricks that come in actual contact with the fire or the heated gases of combustion must be "firebrick"; for the cement that holds them together, "fireclay" is to be used. The common brick and mortar of which the outer walls and settings of boilers are constructed will not answer for the lining, as they are unable to withstand the heat.

78. When rebuilding or repairing a wall or furnace lining in which firebricks are used, fireclay is mixed with sufficient water to make a thin paste, and just enough of it is placed between the bricks to make a joint, the object being to get the bricks as close together as possible.

79. The foregoing instructions apply to both fire-tube and water-tube boilers; a difference exists, however, between the method of cleaning the tubes of each type.

80. Cleaning External Surfaces of Water Tubes.—In the water-tube type the external surfaces of the tubes are cleaned by using a steam hose to blow off the soot that lies on the upper surfaces. When the soot is rather sticky, the steam hose is not as effective as is desired, hence a tool made as shown in Fig. 5 has been devised, by which each and every tube can be thoroughly cleaned off from the top to the bottom row. This tool is simply a piece

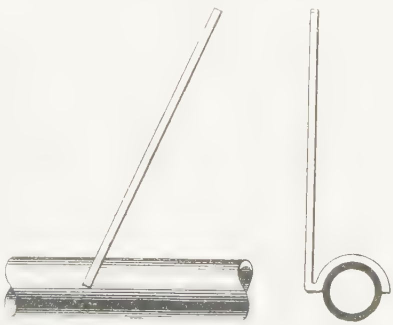


FIG. 5.

of $\frac{1}{2}$-inch square iron bent as shown, and in use is passed down between the vertical rows of tubes and then turned so that it lies upon the tube to be cleaned. By running this tool back and forth a few times over each tube the dirt can be effectually scraped off.

81. Cleaning Internal Surfaces of Fire Tubes.—A steam hose may also be used in the tubes of a fire-tube boiler, blowing the soot through and back into the combustion chamber, from which it can be removed through the door provided for the purpose. Scrapers and wire brushes are also used on this type of boiler for cleaning the internal surfaces of the tubes. These scrapers and wire brushes are usually affixed to a pipe of small size, which is made up of parts coupled together to meet the requirements of space and ease of handling, etc.

82. Keeping Ready for Emergencies. After having cleaned all the external portions of the boiler and made the necessary repairs of brickwork or grates, a new fire may be laid, ready for lighting. The idea of thus early laying a new fire is to be in readiness in case the boiler should be suddenly required, as in the event of one of the others in operation giving out. It is a good plan, as far as possible, to so arrange and divide the work of cleaning and repairs that not too much is under way at one time. This is particularly applicable to the water-tube boiler, with its many plates and adjuncts, where, if all or the greater part were off at one time, there would be considerable delay if the boiler should be hurriedly required. It is for the same reason that all the while the furnaces and back connections, etc. are being cleaned, the water may be allowed to remain in the boiler, thus saving the time required to refill and heat up to the existing temperature, if the boiler is suddenly required. Of course, there may be other considerations of a local or special character that would neutralize or perhaps at least modify the advice just given.

83. Emptying the Boiler.—The furnace and setting having been cleaned and repaired, the boiler should next be

emptied by running the water out through the blow-off. The manhole plates and handhole plates should be taken off and the internal surfaces should be examined with reference to scale and mud.

84. The hose may now be introduced and the boiler thoroughly washed out. If there is much scale and dirt on the surfaces, the boiler should be entered and an attempt made to scrape or otherwise clean those surfaces. At the same time it is advisable to look for broken or loose stays and braces. In some instances special tools have to be made with which the parts may be more readily reached. If the scale is not more than $\frac{1}{32}$ inch thick, it is wise not to make an attempt to remove it, but to let it remain on because it is a real benefit by protecting the surface of the metal from corrosion and pitting.

85. Protective Paints.—A method of preventing internal corrosion and pitting where there is no scale, and to prevent further extension of such if already commenced, is to apply a good coating of paint made of red lead and boiled linseed oil. This paint should be mixed as thick as can be readily applied with a brush to the surfaces. The metal should first be cleaned thoroughly, so that there is no foreign matter between the paint and metal surface to which it is applied.

86. After the paint has been applied it should be allowed to dry hard before filling the boiler with water. This will not take long if the paint is thick, as suggested. It should be understood that *raw* linseed oil must not be used for making the paint. Painting of the internal surfaces of boilers with red lead and boiled oil has been done by many engineers and is successful in a high degree. The paint should be applied frequently, according to the necessity for its use.

87. Another method for the prevention of corrosion and the protection of the internal surfaces is to apply a coating of graphite (plumbago or black lead) mixed in boiled oil and

as stiff as can be applied with a brush. This method also has given good results. There is nothing in any of the materials referred to that can injure the metal, and they can be applied to any type of boiler, excepting a few inaccessible parts of some.

88. Washing Out Water-Tube Boilers.—Water-tube boilers are not as easily washed out with a hose as are boilers of the fire-tube type. The mere fact that the drums can be washed out and that a large quantity of water can be introduced, which finds its way to the bottom and finally out through the blow-off, does not insure that all the tubes are flooded; and even if the water should reach them all, it would not have sufficient force to remove all the mud and loose particles of scale.

89. When in any particular case the time for cleaning water-tube boilers is limited to one or two days, the best that can be done is to remove the manhole plates and flood the interior of the steam drums with water that is entering with some force, either from pressure in the mains from which the supply is drawn or from a pump that receives its supply from a well or some other source. In order that the effect may be as far reaching as possible, it will be necessary to continue the flood for a considerable time—say about 1 hour.

90. The only thorough way of washing a water-tube boiler is to introduce the hose nozzle in each and every tube, and, commencing at the top, to wash out the contents, which fall into the mud drum. This method consumes considerable time, but it pays to do so, provided the time can be spared. Every water-tube boiler should be so treated at least once every 12 months, with the ordinary washing out every month or 6 weeks between.

91. Examining the Attachments.—After the boiler has been washed out, the different attachments should next be examined and overhauled and any repairs necessary should be made. All the valves should be opened and closed to test the practicability of so doing if called on suddenly

while the boiler is under steam. The packing should be renewed in the stuffingboxes of the valve stems whenever that which is in becomes so hard that the gland cannot be screwed up and the stems leak. If any of the valves leak, repair them, if possible; otherwise place in new ones. When plug cocks are used, they should be removed from the shell and cleaned and then rubbed in with oil, turning them completely around several times, so that they be kept true and the contact surfaces smooth. This should be done even if the cocks do not leak, for the reasons given. Leaky cocks must be ground in with ground glass and oil; if a cock leaks badly, it is best to have both the plug and the shell trued in a lathe. In case a repair is not feasible, a new cock should replace the old one. In no case permit a leak to exist when a little time and care will remove such an occurrence.

92. While safety valves do not frequently get out of order (especially those of standard make, because of the excellence of the workmanship thereon), it is desirable to examine them periodically to see that all the parts are intact and not injured to any extent. The proper time to thoroughly overhaul the safety valves, that is, to disconnect and take everything apart, is just before the annual inspection, so that the adjustments may be made by the government inspector or insurance inspector, as required by law or insurance companies. The foregoing refers to places in which license laws are in operation or where the boiler is insured by one of the insurance companies. In places where there is no license law or insurance, the engineer in charge can overhaul the valves when he sees fit and set them to meet the requirements, with due regard to safety.

93. Next in importance to the safety valve is the main stop-valve. This valve should be kept in perfect condition, so that it can be depended on to open and close easily. The packing around the stem should never be allowed to remain so long that it becomes hard, because the steam will then leak out, and in the vain attempt to prevent it by tightening the

gland, the stem is liable to be hugged so tight that it cannot be turned and then may be twisted off in the attempt to move it.

94. When overhauling the stop-valve (or any other valve, in fact), nuts, bolts, and all screwed parts should be placed together with a coating of graphite and cylinder oil on the screw threads, so that they may be taken apart again at some future time without danger of sticking, and, in the case of the bolts, of twisting off. Graphite is particularly mentioned because of its lubricating qualities, whether wet or dry. It should be mixed with oil because it can more readily be applied to the parts in that form.

95. No wrench should ever be used to close any of the valves on the boiler. The wheel on the stem is designed to furnish all leverage that the valve should be closed with. If more than that is applied something will be strained, with perhaps disastrous results. Such treatment has been known not only to bend stems, break disks, strain the threads, but also to actually crack the valve body itself. When a valve cannot be made tight—steam- or pressure-tight—by closing it with the wheel, it is time that an examination be made and the cause removed.

96. If the gauge glass is very dirty, it should be removed and a new one put in. The new glass should be packed with new washers, so that each will adjust itself to the other gradually and thereby relieve the glass of the stress it would otherwise suffer, especially where the fittings are not exactly in line with one another. A few glasses of the required size and length should be kept on hand constantly, because that on the boiler is liable to break at any time; after closing the valves a new glass can be inserted in a few moments. For closing the valves, a rod 8 or 10 feet long, with a forked end that can be placed between the spokes of the wheels, will be found handy and prevent the scalding of the attendant.

97. When inserting a new glass, it is sufficient to screw the gland down with the fingers only, and when the whole attachment is heated to the usual temperature, the packing can again be tightened if necessary. The glands at the top and bottom ends of the glass should be tightened alternately, so that the stress be equally divided.

98. Having overhauled all the larger valves and attachments, the smaller ones should be similarly treated. The feed check- and stop-valves, particularly, should be placed in good order and care should be taken that *all* the pipes are free and clear. If there are no further repairs to be made in addition to the general overhauling, the boiler may be gotten ready for starting.

BOILER REPAIRS.

DIFFERENCE BETWEEN OVERHAULING AND REPAIRS.

99. It frequently happens that in connection with the general overhauling, repairs also have to be made. Overhauling and repairs are not synonymous terms, although both come under the head of maintenance, and in cost accounts connected with a steam plant the distinction should be observed. Repairs generally necessitate the renewal of certain parts that have to be purchased; overhauling consumes only the time occupied in so doing.

REPAIRS OF WATER-TUBE BOILERS.

100. Leaky Tubes.—The parts of a horizontal water-tube boiler that require the most frequent repairs are the tubes. These will occasionally leak at the front end and back end around the part that is expanded into the headers. Sometimes (particularly at the back ends), owing to small unsuspected leaks around the expanded portion, the external surface of the tube for 3 or 4 inches from the header becomes corroded to such an extent that small pinholes soon appear

and leak a small stream of water, which is readily traced to its source. Again, those portions of the tubes that are directly over the fire give out, small holes appearing, which, in this case, are due to pitting on the internal surfaces. Sometimes a tube will give out at the weld, owing to imperfections in the workmanship. This is the most dangerous happening that water-tube boilers are liable to, because of the danger of the tube opening up for quite some distance along the weld when once started. Fortunately, this seldom happens when compared with the number of tubes that are in use.

101. When a leak appears around the expanded portion of a tube, it should be stopped at once by applying a roller tube expander. If such a leak is allowed to continue in an upper tube, it will not only ruin the tube in question, but several others beneath it, because of the corroding action of the constantly dripping water and the gases of combustion.

102. It is difficult to exactly locate a leaky tube in most horizontal water-tube boilers, unless it happens to be in the top or bottom rows, which can be easily reached. Especially is this true where there is more than one tube leaking, the water descending and wetting a number of tubes in its path. In such a case, where doubt exists as to which tube or tubes are leaking, it is better to expand *all* that are in the vicinity of a leak. This action will in the long run save time, and when the expander is handled intelligently no harm whatever is done to those tubes that actually did not leak.

103. When expanding an old tube, care should be taken not to expand too much, especially if the tube is thin, for the action of the expander will make it still thinner; hence, no more than just enough should be given. When the rollers in the expander make a clear and well-defined track in the tube, showing thereby that it is forced equally and evenly against the header, then it is time to discontinue and commence on another. The expander should be oiled

while using; even water has been used for the purpose of lubrication with good results.

104. Tubes that are badly corroded, extremely thin at the expanded ends, or that give out in the portion over the fire should be condemned and withdrawn from the boiler. When it becomes necessary to cut out a tube, care must be taken not to injure the header in which the tube is expanded. The tool used and the manner of using it is shown in Fig. 6,

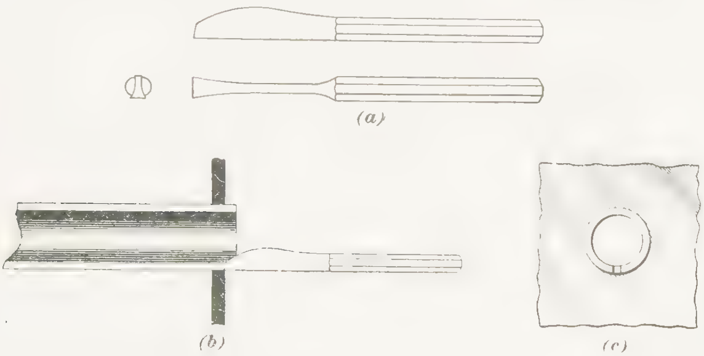


FIG. 6.

in which illustration the cutting tool, known as a **ripper,** is shown at (*a*). Fig. 6 (*b*) shows how it is applied and Fig. 6 (*c*) shows an end view after the tube has been cut. By means of the ripper a slit about $\frac{1}{8}$ inch wide and extending about 1 inch beyond the inside of the sheet is cut; the end of the tube can then readily be squeezed together so that the tube will pass through the hole. With reasonable care there is little danger of cutting into the tube-sheet.

105. Circulating tubes in the water-tube boilers are treated in a similar manner to the tubes when repairs or renewals are necessary.

106. Leaky Piping.—Leaky, corroded, and thin pipes, no matter for what purpose they are used about the boiler, should be replaced with new ones at once.

107. Repairs of Drums.—Owing to the location of the steam drums and mud drums in relation to the fire, they

seldom require repairs, especially if the boiler had been well built at first. However, should any of the plates, seams, or units give out, the method of repairing will be the same in general as that described in repairs to fire-tube boilers.

REPAIR OF FIRE-TUBE BOILERS.

108. Leaky Tubes.—There is no difficulty in locating a leaking tube in the fire-tube type of boiler; if a tube leaks at either end, it is readily seen, because the ends are exposed to view. Even should a leak occur at some place remote from either tube-sheet, the tube in question can easily be found and the usual remedy applied.

109. If it be desired to save a tube whose ends are leaking—a leak that the expander does not repair—and that is otherwise in good condition, sleeves or ferrules may be inserted, as described in Art. **65.**

110. If the tube is too far gone, it should be cut out and a new tube put in. Sometimes a tube gives out in such a manner and at such a time that neither repairs nor renewal can be accomplished as before described; then a tube stopper may be used until opportunity offers to put in a new tube. This will prevent the water from leaking out, but at the expense of entirely losing the amount of heating surface of the tube. In case of such emergency, it is of more importance to plug the tube than to consider the temporary loss of heating surface. Occasions may arise where it may be necessary to place stoppers in two, or perhaps more, tubes.

111. Tube Stoppers.—A home-made tube stopper is shown in Fig. 7. It consists of two tapering pine plugs

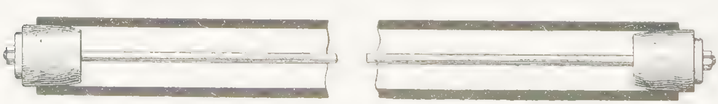


FIG. 7.

having a central hole through which a rod made of $\frac{1}{2}$-inch round iron is passed. This rod is provided with

nuts and washers at each end, by means of which the plugs can be drawn home. This tube stopper is quite effectual as far as stopping the leak is concerned, but is open to the objection that a man must enter the back connection in order to place the plug in the rear end in position and in order to put the nut and washer on the rod.

112. The objectionable feature of the tube stopper shown in Fig. 7 is overcome by the design shown in Fig. 8,

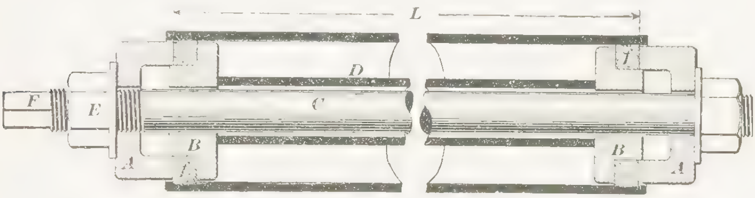


FIG. 8

which has long been used in marine work. Cast-iron washers *A* and *B* having a rubber ring *f* about $\frac{1}{2}$ inch thick between them are placed on each end of a tie-bolt *C* having nuts and washers on each end; a sleeve *D* made of gas pipe is placed between the two pairs of washers. The distance *L* is made about 1 inch less than the length of the tube. To stop a leaky tube, the stopper is inserted from the front and the nut *E* is tightened, while the tie-bolt is prevented from turning by a wrench put on the square end *F*. Tightening the nut *E* draws the washers *A* and *B* together, compressing the rubber rings, and thus forcing them against the inside of the leaky tube. This kind of a tube stopper can readily be applied without stopping the operation of the boiler.

113. Leaky Seams and Rivets.—Leakage is not confined to the tubes only in the fire-tube type of boiler, but may and frequently does occur at the seams and around the rivets, especially those that are in contact with, or in close proximity to, the fire. A leak may occur at such places and not be seen while the boiler is steaming, because of the rapid evaporation of the leaking water by the fire, unless the

leak be so large as to resist immediate evaporation. After the boiler has been cut out of service and the fire drawn out, the presence of such a leak will be revealed by the presence of some deposit on the metal where the leak is. Even should there be no deposit whatever, there certainly will be some distinguishing mark by which the presence of a leak is indicated. The fact that no such mark, or deposit, can be found near a seam is pretty clear evidence that the seam is tight.

114. A leaky seam is made tight by calking; that is, the seam is closed by driving a round-nosed calking tool against it. It should be remembered that repeated light blows of the hammer on the tool are more effectual than heavy blows; in fact, heavy blows are liable to cause the seam to leak worse than at first.

115. Calking also stops leakage around rivet heads, provided the rivet has been properly driven at first. If a rivet has been forced into a mismatched hole and then starts to leak, calking will seldom make it tight; the only real remedy is to cut the rivet out, clean out the hole, and drive in a new rivet of the next larger size. This will, of course, slightly weaken the plates, but not to such an extent as to preclude this plan of action, which certainly will leave the joint in better condition in every respect than it was before.

116. Cracks.—If a crack should appear in any of the plates, it may be repaired by drilling a small hole at each end of the crack, calking the crack, and putting a patch over it. The idea of drilling the holes at the ends is to prevent the crack from extending further, as it would otherwise do; if the crack should be extensive, the part of the plate containing it should be cut out and a patch put on.

117. Patching.—There are two kinds of patches used on boiler repairs, viz: *soft patches* and *hard patches*. A **soft patch** is one that is put on with bolts and nuts or with tap bolts; red-lead putty is used between the plates to make a water-tight joint under pressure. Soft patches at best

are only makeshifts and should not be used at all on plates that are subjected to the direct heat of the fire.

118. Hard patches are secured with rivets and should be the only kind of patches used on boiler work. In putting hard patches on plates that are directly over the fire, the weak or otherwise affected part of the plate should be cut out before the patch is placed on. The patch, whether hard or soft, should be placed on the inside, that is, the water side of the plate. This is done for the same reason that manhole and handhole plates are placed on the inside instead of on the outside of the boiler. Plates that are badly blistered or otherwise injured should be cut out and replaced with new plates.

119. Laminations.—Sometimes a lamination in a plate is revealed by the action of the fire, causing a bag or blister to appear. Laminations are due to slag and other impurities in the metal, which in rolling the plates become flattened out, as shown at *a*, *a* in Fig. 9. Under the action of the

FIG. 9.

heat the part exposed to the fire will form a blister, as shown in the figure, which may finally open at the point *b* or *c*, depending on the position of the slag in the plate. The laminated portion of the plate may be but very small; in that case a hard patch can be put on. If there are a number of laminations in the same plate, it is advisable to put in a new plate.

120. When a lamination or an otherwise affected portion of a plate has to be cut out, the form of the piece cut out should be as nearly circular as possible. In any case no sharp corners should be left, because of the tendency of cracks to start at such places.

121. Extensive Repairs.—Extensive repairs to boilers should always be made by experienced boilermakers that have the necessary equipment for such work. The lesser repairs may be done by the engineers themselves, such, for example, as expanding tubes, cutting out and putting in tubes and sleeves, etc. But where the cutting out of sheets or portion of sheets that are to be patched, the driving of rivets, and similar work is to be done, it is generally advisable that none but a boilermaker should attempt it.

INSPECTION.

OCULAR INSPECTION.

122. Inspection of boilers is one of the most important duties of an engineer, because on this depends largely their safety and good condition. Owing to the several deteriorating agents that tend to weaken and shorten the life of a boiler if their effects are neglected, it is important that inspections be held periodically and that notes be made of the general condition from time to time. An engineer should not depend entirely on the report of a government or insurance inspector upon the condition of his boiler, but should make inspections himself and actually see the condition of the boiler, the idea being that very valuable knowledge can thus be gained.

123. Every part, both external and internal, should be thoroughly examined. Corrosion and its progress should be noted and action taken to entirely stop it or at least to limit it in extent. Leaks should be looked for and stopped immediately. The internal surfaces of the plates should be examined at the water-line for pitting and at the junction of the plates in the seams for grooving. The condition of stays and braces and whether they are tight should be noted. Should staying be found loose, it must be tightened by whatever means are available.

HAMMER TEST.

124. The ocular inspection should be accompanied by striking the plates, stays, and tubes with a hammer to determine their soundness; this is called a **hammer test.** Sound plates and tubes when struck with a hammer emit a clear, bell-like ring, while those that are thin or defective give forth a dull sound, similar to that of a cracked piece of pottery. A broken stay gives a peculiar sound that cannot be described by words.

HYDROSTATIC TEST.

125. Value of Hydrostatic Test.—The **hydrostatic test** (filling the boiler with water and applying a pressure by means of a pump or otherwise) is valuable only in showing leaks and to determine the ability of the boiler to withstand a prescribed pressure. It will not reveal weak places unless such places are so weak as not to be able to stand the required pressure. But it frequently happens that thin places do stand the pressure to an astonishing degree, although they are in a dangerous condition, hence the hydrostatic test should always be supplemented by an ocular inspection and a hammer test.

126. Objections.—There is an objection against the hydrostatic test that there is danger of straining the plates beyond the elastic limit and that thereby a boiler may be permanently injured which would have been safe at the working steam pressure. The insurance companies in most cases depend on the hammer test and ocular inspection, but use the hydrostatic test for new boilers, old boilers extensively repaired, and all boilers that cannot be examined thoroughly inside and outside.

127. Precautions. — When applying the hydrostatic test, the escape of air from the boiler while filling with water should be provided for, leaving some valve or cock open until the water is forced out in a solid stream. The valve or other opening used for the escape of air must be located as high as possible, so that practically little or no

air remains in the boiler when it is closed. This precaution is necessary when it is considered that should the boiler burst under pressure while still containing air, the parts, by reason of the expansion of the air, are liable to fly with great force, perhaps injuring some one in their flight. A boiler from which all air has escaped and bursting under the hydrostatic test will not do any serious damage.

128. When there are two or more boilers connected by piping, the intercommunication being broken only by a valve, it will be necessary to place a blank flange between the valve and the boiler that is to be tested, thus completely isolating the boiler from those in operation. This is done as a measure of safety, which the valve alone is not capable of insuring.

129. In making the hydrostatic test, the pressure must be applied very slowly and carefully and the gauge must be watched for any drop of pressure that would denote a yielding of some part of the boiler. New boilers are tested by hydrostatic pressure to reveal leaky joints or rivets. When the seams or rivets are not tight, water trickles out in drops or spins out in a stream. Such places are marked with chalk and afterwards recalked. Boilers are usually tested hydrostatically to $1\frac{1}{2}$ times the pressure they are to carry.

130. Testing by Heating the Water.—A method of applying the hydrostatic test that is used by many engineers is to fill the boiler full of cold water and build a gentle fire in the furnace. As the temperature of the water rises, it expands and thus subjects the boiler to pressure. It is urged in favor of this method that the pressure is raised steadily and the boiler is not as liable to be injured as it is when subjected to sudden and jerky rises of pressure due to the working of a pump. The temperature of the water should in no case be made to rise above 212°, since, if a rupture should take place, the pressure of the water would lower to that of the atmosphere, and the temperature of

the water being above the boiling point at atmospheric pressure, a quantity of the water might suddenly become steam and cause an explosion.

131. The inspection of steam boilers should begin at the place where the plates are manufactured and continue as long as the boiler is in use.

BOILER EXPLOSIONS.

CAUSE.

132. A **boiler explosion** can be caused only by overpressure of steam. Either the boiler is not strong enough to carry its ordinary working pressure or else for some reason the pressure has been allowed to rise above the usual point.

133. In the first case, the boiler may be too weak for the working pressure, because: (1) It is poorly designed. (2) The material or the workmanship may be poor. (3) The boiler may have become weakened by corrosion. (4) The boiler may have been weakened by careless or reckless management, such as letting cold water come in contact with hot plates, or blowing the boiler off hot and then quickly filling it with cold water.

134. When the pressure rises above the usual point that the safety valve is supposed to be set for, the fault is probably due to the sticking or overweighting of the safety valve. Some very disastrous explosions have been caused by closing a stop-valve between the safety valve and boiler while cleaning the latter and then forgetting to open the stop-valve. It cannot be too strongly urged that a stop-valve should never be placed between the safety valve and boiler.

135. Low water may cause explosions in internally fired boilers, but will rarely cause externally fired boilers to explode, although low water is often assigned as the cause.

PREVENTION.

136. Explosions may be prevented by observing the following directions:

1. Have the boiler inspected or tested to determine its safe working pressure.

2. Use all possible care to prevent internal and external corrosion, and be careful that the plates do not become reduced to an excessive thinness without your knowledge.

3. Do not strain the shell by subjecting it to great changes in temperature; that is, do not blow it off hot and quickly fill up with cold water; do not deluge red-hot plates with cold water; and do not let in more cold air through the furnace door than necessary.

4. Do not overload the safety valve to make the task of firing easier, and do not let it become corroded fast to its seat.

5. Do not allow the water to get very low.

6. Cases have been known where the sudden opening or closing of a large stop-valve leading to the main steam pipe has led to an explosion. There is considerable risk in so doing, hence it is well to open or close such a valve slowly and cautiously.

7. Do not try to use a boiler after it is worn out. Replace it with a new one.

8. When boilers are placed in buildings containing a great number of people, use a type of boiler in which the danger of a disastrous explosion is reduced to a minimum.

BOILER TRIALS.

BOILER-TRIAL DETAILS.

PURPOSES OF A BOILER TRIAL.

1. A **boiler trial,** or **boiler test** as it is often called, may be made for one or more of several purposes, the method of conducting the trial depending, naturally, largely on its purpose. The boiler trial may vary from the simplest one, in which the only observations are the fuel burned and the water fed to the boiler in a stated period of time, to the elaborate standard boiler trial, in which special apparatus and several skilled observers are essential.

2. The purposes for which a boiler trial is made are as follows:

1. To determine the steam-making value of a fuel, as measured by pounds of water evaporated per pound of fuel or per dollar.

2. To determine the standard horsepower of a boiler, according to the rating of the American Society of Mechanical Engineers.

3. To determine the efficiency of the boiler; that is, to determine what percentage of the heat liberated by the fuel is usefully expended in evaporating water into dry steam.

4. In conjunction with an engine test, to determine the coal consumption and steam consumption in pounds per indicated horsepower per hour.

COMPARATIVE FUEL VALUE BOILER TRIAL.

3. Introduction.—In the management of a steam plant the successful engineer will never lose sight of the commercial features of the management, since the cases in which the running expenses are of no consequence are few and far between. Since fuel, chiefly in the form of coal, is one of the most important items of running expenses, any reduction in the fuel cost represents a direct saving that is likely to be larger than any saving effected in one of the less important items.

4. Finding the Commercial Value of a Fuel.—The commercial value of a given fuel for an existing set of conditions can be determined only by an extended trial, determining, from carefully kept records, how much water is evaporated per dollar, and adding to the total fuel cost the cost of attendance and the cost of removal of ashes. Then, if a careful record be kept of the same items under the same conditions, but with different fuels, the commercial values of these fuels *in relation to the boiler plant where the test is made*, can be established, and the cheapest fuel for that plant can be selected.

5. It will have been noticed that the basis of comparison is not the water evaporated per pound of fuel, but the water evaporated per *dollar*. This basis of comparison is chosen for the simple reason that the owner of a boiler plant is not concerned about a high evaporative power of a fuel, but rather about the fuel that will run the plant at the least expense.

Since it may not be apparent why the cost of attendance and removal of ashes should be considered, it may be stated that when low-priced fuels are used, it may be necessary to have more help, since generally a larger quantity must be burned. Likewise, with a low-grade (cheap) fuel there will be much more ashes to be removed. The item of wear and tear is generally omitted, since it is usually a constant quantity, irrespective of the fuel used.

6. Keeping the Records.—A convenient method of keeping the record is shown below.

RECORD OF COMPARATIVE FUEL TESTS.

Kind of Fuel.	Amount Burned. Tons of 2,000 Pounds.	Cost Per Ton, in Dollars. Delivered.	Total Fuel Cost. Dollars.	Attendance. Dollars.	Ash Removal. Dollars.	Water Evaporated. Pounds.	Water Evaporated Per Dollar. Pounds.	Water Evaporated Per Pound of Fuel. Pounds.
Pea	52	1.75	91.00	75.00	3.20	524,000	3,097	5.04
Buckwheat...	75	1.00	75.00	75.00	4.00	490,000	3,182	3.27
Nut	49	2.50	122.50	75.00	2.40	543,000	2,716	5.54
Egg	35	3.50	122.50	75.00	1.60	560,000	2,813	8.00
Stove........	43	2.90	124.70	75.00	2.00	552,000	2,737	6.42

7. Analyzing the Records.—In this record the water evaporated per dollar is found by dividing the water evaporation in pounds by the sum of the total fuel cost, the cost of attendance, and the cost of removing ashes. The evaporation per pound of fuel is found by dividing the total evaporation in pounds by the total weight of fuel in pounds. Analyzing the record, it is seen that the order of evaporative performance per pound of fuel is egg, stove, nut, pea, and buckwheat, the egg coal evaporating nearly $2\frac{1}{2}$ times as much water as the buckwheat. Comparing their economic performances, however, from the standpoint of water evaporated per dollar, the order is as follows: buckwheat, pea, egg,

stove, and nut coal. The analysis shows that for the particular boiler plant in which the tests were made, buckwheat coal, although very low in evaporative power, is the cheapest.

8. Precautions.—The errors of concluding that a very cheap fuel must be necessarily cheap in the long run, and conversely, that a coal high in evaporative power is on that account the best coal, must be guarded against. The most satisfactory way in which a correct conclusion as to the respective commercial values of different fuels can be arrived at is to make an actual test and then compare their performances in the manner shown.

9. Making the Test.—The making of a test for the purpose of finding the comparative value of different fuels for a given boiler plant is ordinarily a very simple matter, especially so in a plant that is running steadily under a uniform load; that is, where the demand for steam is practically constant throughout the day and the same day after day.

10. In order that the slight variations of conditions existing in the plant just mentioned may average themselves, it is advisable to let the test of each fuel extend over quite a period, say not less than a week. There is no particular necessity that the duration of the tests of each fuel should be alike, that is, one coal may be used for a week and another may be used for a month; it is absolutely necessary, however, to keep an accurate record of the coal used and water evaporated. If it is convenient to make the duration of all tests alike, it may be done merely in order that no suspicion of unfairness may be created, but, as previously stated, there is no real necessity for this.

11. The making of a comparative fuel value test in a plant running intermittently does not require any essentially different method of procedure from that followed in a steady-running plant. The only precaution to be observed is to let the test of each fuel extend over a considerable period of time, in order that conditions may be fairly uniform for each fuel.

12. Measuring the Coal.—The question of whether the coal fed to the boilers should be weighed or not depends on circumstances. In most cases, and especially where a *known* weight of coal sufficient for at least a week's run or more is at hand and circumstances permit the test of this fuel to be continued until it is all burned, it would scarcely be necessary. The only precaution to be observed is to find out whether the fuel was paid for by the long ton (2,240 pounds) or by the short ton (2,000 pounds), in order that the weight may be correctly reduced to pounds for the record, in case it is desired to find the water evaporated per pound of fuel.

13. Where storage capacity is so small that coal must be delivered at very frequent intervals, it would be advisable to weigh out the coal as used. Incidentally it may be remarked that this plan gives a good check as to the actual amount of coal delivered. A convenient plan for weighing the coal is to have a box with one side open placed on a platform scale. A weight is placed on the scale beam just sufficient to balance the empty box. The scale is then set at 400, 500, or 600 pounds, as desired. The coal is shoveled into the box until the beam rises, so that the box contains just 400, 500, or 600 pounds, as the case may be. The coal is then fed from the box into the furnace.

A plan often followed is to weigh one wheelbarrow load of coal and then afterwards simply fill the wheelbarrow with the same amount of coal, as nearly as can be judged. This plan is simple, but liable to be very inaccurate, and consequently is not to be recommended.

14. Measuring the Water.—The amount of water evaporated may in a test for comparative fuel values be taken as equal to the amount of feedwater supplied without introducing any serious error. The most reliable method of measuring the feedwater delivered to the boilers is to weigh it. A convenient way of doing this is to have two tanks *A* and *B*, Fig. 1, one above the other. The supply of water is fed through the pipe *C* into the upper tank *A*, which rests on a platform scale *D*. After balancing the tank, the scale

may be set to weigh 500 or 600 pounds of water, the water being run in until the beam rises and then shut off. The

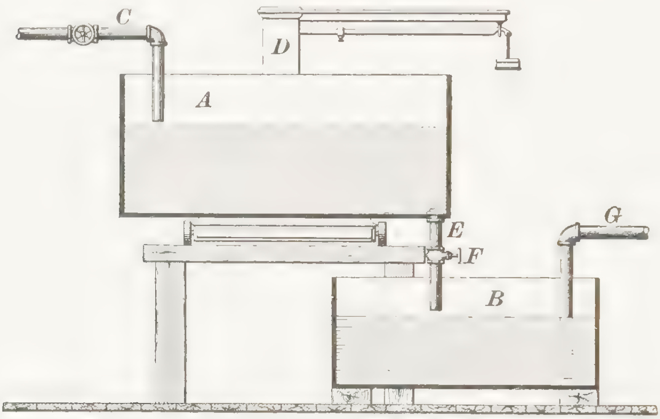


FIG. 1.

upper tank is provided with a pipe *E* and valve *F*, by means of which the water may be discharged into the lower tank, from which it is fed to the boiler through the pipe *G*.

15. The attendant who keeps the record of coal supply or water supply should become accustomed to making the tally on his blank just before or after some regular operation. For example, the person who weighs the feedwater should record each tankful, say immediately after closing the valve in the supply pipe, or perhaps after emptying the upper tank into the lower. If this precaution is not observed, the attendant is liable to become uncertain as to whether he has recorded the previous data, and a mistake is almost certain to result.

16. In some cases it is more convenient to use a water meter to measure the feedwater than to weigh it. A good water meter will register correctly within 1 per cent. of the actual amount, and the amount of error is practically constant, so that if the amount of error be known, a correction in the reading of the meter can readily be made. While the testing of a meter to determine its constant error is a

comparatively simple matter, it requires facilities rarely to be found in a steam plant, and hence the best plan is to have it tested by the manufacturer. When a water meter is used, it should be attached directly to the feedpipe; a meter attached to a main water pipe should never be considered in a boiler test for the obvious reason that it shows how much water is delivered to the plant, but not how much water is fed to the boilers. Water meters almost invariably register the amount of water passing through them in gallons; their indication may be readily reduced to pounds by multiplying by 8.355, which is the number of pounds of water contained in the United States (Winchester) gallon of 231 cubic inches capacity.

HORSEPOWER AND EFFICIENCY TESTS.

17. Finding the Standard Boiler Horsepower.—When making a horsepower or efficiency test, a more elaborate method of procedure is required than for a comparative fuel value test. The reason for this will become apparent when it is considered that different boilers generate steam at different pressures, different feedwater temperatures, and different degrees of dryness; this being the case, it follows that before the standard horsepower of boilers can be found, the evaporation must be reduced to a basis common to all, which is the **equivalent evaporation from and at 212°**. The standard horsepower is then obtained by dividing the equivalent evaporation by $34\frac{1}{2}$, which represents the number of pounds of water evaporated from and at 212° that has been agreed on by the American Society of Mechanical Engineers as the unit of standard boiler horsepower.

18. Finding the Efficiency.—The **efficiency** of a boiler may be defined as the ratio of the heat utilized in evaporating water to the total heat supplied by the fuel. The efficiency thus calculated is really the combined efficiency of the furnace and boiler, as it is not easily possible to separately determine the efficiency of each.

19. The amount of heat supplied is determined by first accurately weighing the fuel used during the test and deducting all the ash and unconsumed portions. This weight in pounds is multiplied by the total heat of combustion of a pound of the fuel, as determined by an analysis, the product being the total number of heat units supplied during the test under the assumption that combustion was perfect. The heat usefully expended in evaporating water is obtained by first weighing the feedwater and correcting this weight according to the quality of the steam; the corrected weight is then multiplied by the number of heat units required to change water at the temperature of the feed into steam at the observed pressure.

Rule 1.—*To find the efficiency of a boiler, expressed in per cent., divide 100 times the number of heat units usefully expended in evaporating water by the number of heat units supplied by the fuel.*

Or, let E = efficiency of boiler;
A = heat units usefully expended;
B = heat units applied.

Then, $$E = \frac{100\,A}{B}.$$

Example.—A boiler trial shows a useful expenditure of 186,429,030 B. T. U. and a total supply of 270,187,000 B. T. U.; what is the efficiency of the boiler plant?

Solution.—Applying rule **1**, we have

$$E = \frac{100 \times 186{,}429{,}030}{270{,}187{,}000} = 69 \text{ per cent. Ans.}$$

FINDING THE EQUIVALENT EVAPORATION.

20. The actual evaporation of a boiler is rarely equal to the amount of water pumped into the boiler, since a boiler seldom, if ever, will produce dry saturated steam. For this reason the amount of feedwater must be multiplied by a factor representing the amount of dry steam contained in

every pound of water apparently evaporated. The manner in which this factor is determined will be explained further on.

21. The equivalent evaporation is readily determined by the following rule:

Rule 2.—*To reduce actual evaporation to equivalent evaporation, subtract the observed temperature of the feedwater from the total heat of 1 pound of steam above 32° at the pressure of evaporation. Add 32 to the remainder and multiply the sum by the actual evaporation in pounds. Divide the product by 966.1.*

Or, let W = actual evaporation;
H = total heat of steam above 32° at observed pressure of evaporation;
t = observed feedwater temperature;
W_1 = equivalent evaporation from and at 212° F.

Then.
$$W_1 = \frac{W(H - t + 32)}{966.1}.$$

Example.—A boiler generates 2,200 pounds of dry steam per hour at a pressure of 120 pounds, gauge. The temperature of the feedwater being 70°, what is the equivalent evaporation?

Solution.—According to the Steam Table, the total heat H corresponding to a gauge pressure of 120 pounds is 1,188.64 B. T. U. Applying rule **2**, we get

$$W_1 = \frac{2{,}200 \times (1{,}188.64 - 70 + 32)}{966.1} = 2{,}620 \text{ lb., nearly. Ans.}$$

22. In rule **2** the quantity $\frac{H - t + 32}{966.1}$ that changes the actual evaporation of a pound of water to equivalent evaporation from and at 212° is called the **factor of evaporation.** To facilitate calculation of the equivalent evaporation, the following table of Factors of Evaporation is inserted.

The equivalent evaporation is found by multiplying the actual evaporation by the factor of evaporation taken from the table.

FACTORS OF EVAPORATION.

Temperature of Feed-water.	Gauge Pressures.															
	25	30	35	40	45	50	60	70	80	90	100	120	140	160	180	200
	Factors of Evaporation.															
32	1.204	1.206	1.209	1.211	1.212	1.214	1.217	1.219	1.222	1.224	1.227	1.231	1.234	1.237	1.239	1.241
40	1.196	1.198	1.201	1.203	1.204	1.206	1.209	1.211	1.214	1.216	1.219	1.223	1.226	1.229	1.231	1.233
50	1.185	1.187	1.190	1.192	1.193	1.195	1.198	1.200	1.203	1.205	1.208	1.212	1.215	1.218	1.220	1.222
60	1.175	1.177	1.180	1.182	1.183	1.185	1.188	1.190	1.193	1.195	1.198	1.202	1.205	1.208	1.210	1.212
70	1.165	1.167	1.170	1.172	1.173	1.175	1.178	1.180	1.183	1.185	1.188	1.192	1.195	1.198	1.200	1.202
80	1.154	1.156	1.159	1.161	1.162	1.164	1.167	1.169	1.172	1.174	1.177	1.181	1.184	1.187	1.189	1.191
90	1.144	1.146	1.149	1.151	1.152	1.154	1.157	1.159	1.162	1.164	1.167	1.171	1.174	1.177	1.179	1.181
100	1.134	1.136	1.139	1.141	1.142	1.144	1.147	1.149	1.152	1.154	1.157	1.161	1.164	1.167	1.169	1.171
110	1.123	1.125	1.128	1.130	1.131	1.133	1.136	1.138	1.141	1.143	1.146	1.150	1.153	1.156	1.158	1.160
120	1.113	1.115	1.118	1.120	1.121	1.123	1.126	1.128	1.131	1.133	1.136	1.140	1.143	1.146	1.148	1.150
130	1.102	1.104	1.107	1.109	1.110	1.112	1.115	1.117	1.120	1.122	1.125	1.129	1.132	1.135	1.137	1.139
140	1.092	1.094	1.097	1.099	1.100	1.102	1.105	1.107	1.110	1.112	1.115	1.119	1.122	1.125	1.127	1.129
150	1.082	1.084	1.087	1.089	1.090	1.092	1.095	1.097	1.100	1.102	1.105	1.109	1.112	1.115	1.117	1.119
160	1.071	1.073	1.076	1.078	1.079	1.081	1.084	1.086	1.089	1.091	1.094	1.098	1.101	1.104	1.106	1.108
170	1.061	1.063	1.066	1.068	1.069	1.071	1.074	1.076	1.079	1.081	1.084	1.088	1.091	1.094	1.096	1.098
180	1.050	1.052	1.055	1.057	1.058	1.060	1.063	1.065	1.068	1.070	1.073	1.077	1.080	1.083	1.085	1.087
190	1.040	1.042	1.045	1.047	1.048	1.050	1.053	1.055	1.058	1.060	1.063	1.067	1.070	1.073	1.075	1.077
200	1.030	1.032	1.035	1.037	1.038	1.040	1.043	1.045	1.048	1.050	1.053	1.057	1.060	1.063	1.065	1.067
210	1.020	1.022	1.025	1.027	1.028	1.030	1.033	1.035	1.038	1.040	1.043	1.047	1.050	1.053	1.055	1.057

EXAMPLE.—A boiler takes in 4,400 pounds of water at 50° per hour. The steam pressure is 80 pounds, gauge. How much water would this boiler evaporate per hour from and at 212° ?

SOLUTION.—In the table of Factors of Evaporation, opposite a feed temperature of 50° and boiler pressure of 80 pounds, we find the factor 1.203. Multiply the quantity of water actually evaporated, viz., 4,400 pounds, by this factor. $4,400 \times 1.203 = 5,293.2$ lb., the water that would be evaporated per hour from and at 212°. Ans.

23. When neither the feedwater temperature nor the steam pressure appears in the table, the factor of evaporation can be found by interpolation, the method being shown best by an example.

ILLUSTRATIVE EXAMPLE.—What is the factor of evaporation when the feedwater temperature is 122° and the gauge pressure 72 pounds?

SOLUTION.—In the table of Factors of Evaporation, under the column headed 70 and opposite 120 in the left-hand column, is found 1.128; in column headed 80 and opposite 120 is found 1.131. The difference is $1.131 - 1.128 = .003$. In the same vertical columns and opposite 130 are found 1.117 and 1.120, giving a difference of .003. Hence, for an increase of 10 pounds in the gauge reading there is an increase of .003 in the factor of evaporation, or an increase of $\frac{.003}{10} = .0003$ for 1 pound, and of $.0003 \times 2 = .0006$ for 2 pounds, or the amount by which the given steam pressure varies from the next lower tabular pressure. Therefore, for a feedwater temperature of 120° and 72 pounds gauge pressure, the factor of evaporation is $1.128 + .0006 = 1.1286$.

The factor of evaporation must now be corrected for the feedwater temperature. The difference between the numbers opposite 120 and 130 in the two columns headed 70 and 80 is $1.128 - 1.117 = .011$, and $1.131 - 1.120 = .011$, showing that for an increase of feedwater temperature of 10° there is a decrease of .011 in the factor, and for 1° of $\frac{.011}{10} = .0011$, and for 2° of .0022. Hence, the value of the factor of evaporation for a temperature of 122° and a gauge pressure of 72 pounds is $1.1286 - .0022 = 1.1264$ to four decimal places. Ans.

FINDING THE QUALITY OF THE STEAM.

24. Introduction.—In making a boiler trial, it is important to determine as closely as possible how much moisture, if any, the steam contains. Many boilers, especially when

generating steam rapidly, furnish wet steam. The water is carried along with the steam in the form of spray or even in drops. Of course, this water is not evaporated, and if not taken account of, would show for the boiler a higher efficiency than it really possessed.

By the expression "quality of steam," is meant the percentage of the water *fed into* the boiler that is evaporated into pure dry steam. For example, suppose that for every 100 pounds of water fed to the boiler 98 pounds are changed to dry steam and 2 pounds are carried over in the form of water. Then, the quality of steam is 98 per cent. and the percentage of moisture is 2 per cent.

25. Barrel Calorimeter.—It is a rather difficult matter to make a very exact determination of the moisture contained in steam. The apparatus or instrument used for this purpose is called a **calorimeter.** There are many more or less complicated calorimeters in use; about the simplest and most available one for general use is the so-called **barrel calorimeter.** (See Fig. 2.) A barrel or tank *T* holding 400 or 500 pounds of water is placed on a platform scale *S*, filled with water, and weighed. The temperature of the water is registered by a thermometer inserted in the side of the barrel. Steam from the boiler is led through a pipe or hose *U* into the barrel until the temperature of the water reaches 130° to 140° F. The steam is then turned off and the barrel and contents are again weighed. The difference between this weight and the original weight is the weight of the steam led in from the boiler. The average steam pressure throughout

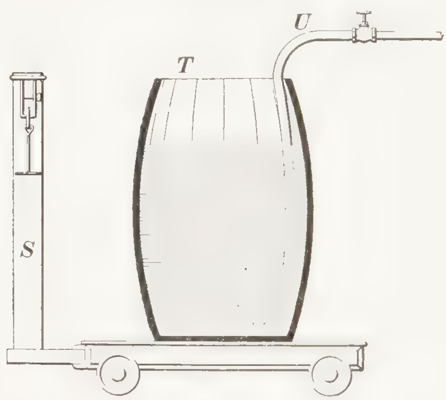


FIG. 2.

the observation must be observed. It is well to have the tube bent as shown in the figure.

26. We know the weight of the cold water and the rise in temperature; we also know the weight of the steam or the mixed steam and water that is led in from the boiler. From the Steam Table, the temperature of the steam can be found, since the pressure is known.

Now, if dry steam comes through the pipe U, the condensation of this steam should raise the temperature of the cold water in the barrel a certain amount. If the temperature is not raised that much, it must be because some of the mixture led into the barrel was water. Very rarely the temperature of the cold water might rise more than it would if the steam were dry, thus indicating that the steam is *superheated.*

Rule 3.—*Multiply the rise in temperature of the cold water by the weight of cold water and divide the product by the weight of the mixture led in through the hose. Subtract the difference between the temperature of the steam and the final temperature of the water from the number first obtained. Divide this final remainder by the latent heat of a pound of steam at the observed pressure. The result will be the quality of the steam.*

Let W = weight of cold water in barrel;
w = weight of mixture run into the barrel;
t = temperature of steam corresponding to the observed pressure;
t_1 = original temperature of cold water;
t_2 = temperature of cold water after steam is condensed;
L = latent heat of a pound of steam at the observed pressure;
Q = quality of steam.

Then,
$$Q = \frac{\dfrac{W(t_2 - t_1)}{w} - (t - t_2)}{L}.$$

Example.—In a calorimetric test, the weight of cold water was 420 pounds; of steam condensed, 36 pounds. The initial temperature of cold water was 40° F., the final temperature was 130° F., and the steam pressure was 60 pounds. Find the quality of the steam.

Solution.—Absolute pressure = 60 + 14.7 = 74.7 pounds per square inch. Latent heat of steam at this pressure is (see Steam Table) 898.5. The temperature of steam at this pressure is 307.2°. Hence, by rule **3,**

$$Q = \frac{\frac{420\,(130 - 40)}{36} - (307.2 - 130)}{898.5} = .9714 = 97.14 \text{ per cent.};$$

that is, the boiler generates a mixture that is composed of 97.14 per cent. dry steam and 2.86 per cent. water. Ans.

27. If Q is greater than 1, it shows that the steam is superheated. The number of degrees of superheat is then found by the following rule:

Rule 4.—*Subtract 1 from the quality of the steam calculated by rule* **3** *and multiply by the latent heat of the steam at the observed pressure. Divide the product by .48.*

Or,
$$S = \frac{(Q - 1)\,L}{.48},$$

where S = superheat in degrees Fahrenheit and the other letters have the same meaning as in rule **3.**

28. The barrel calorimeter must be used very carefully in order to obtain accurate results. The operation should be repeated once or twice before the actual test is made, in order to warm up the barrel. The most important observation is the temperature. This should be taken by a thermometer graduated to fifths or tenths of a degree. The weights should be as accurate as possible. The chief merit of the barrel calorimeter is its availability. It can be rigged up in almost any situation.

29. Separator Calorimeter.—While the barrel calorimeter is probably the most available form of calorimeter, some doubt attaches to the results obtained by its use, and it has been the aim of some noted engineers to design a form of instrument for measuring the quality of the steam that would combine simplicity with reliability and be more trustworthy than the barrel calorimeter.

30. Fig. 3 shows the so-called **separator calorimeter** designed by Professor R. C. Carpenter, Cornell University. The steam to be tested passes through the pipe *a* and head *b* into the mechanical separator *c*. The steam escapes through a series of fine holes and passes over the edge of the cup *d* into the outer space *e*. It passes out, thence, through the shank *f* and hose *g* to a condenser *h*, where it is condensed. The sudden change of direction of the flow of the steam in passing through the fine orifices in *c* causes the water that is suspended in the steam to be thrown into the cup *d*. The quantity thus caught in the cup is shown by the gauge glass *i*. The scale *j* is so graduated that each division indicates $\frac{1}{100}$ pound. The separator frees the steam from entrained water in a very perfect manner, so that only practically dry steam passes to the condenser.

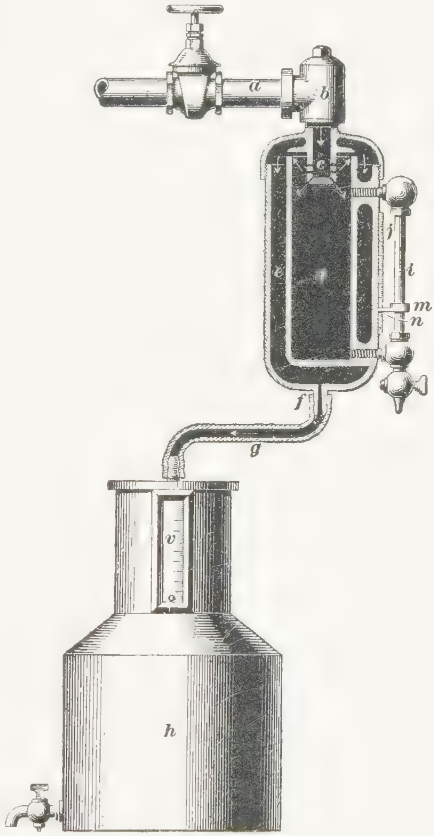


FIG. 3.

The rate of flow of steam is limited by the size of the orifice in *f*, which is usually $\frac{1}{16}$ inch. Condensation with steam at 60 pounds gauge pressure will occur at the rate of about 1 pound in 3 minutes. The condenser should contain from 75 to 80 pounds of water, which should be as cold as

can conveniently be obtained and should fill the condenser to or a little above the zero mark of the scale v. Each division of the scale v indicates $\frac{1}{10}$ pound. As the steam condenses the water level will rise, and the difference between the successive readings of the scale indicates the weight of dry steam actually condensed. Readings should be taken simultaneously upon the scales v and j at the beginning and end of each test. The weight of steam actually tested is the sum of the weights of the steam condensed in h and the water that is caught in d.

31. To use the instrument it must be warmed up slowly, to avoid breaking the gauge glass. The full pressure of steam should then be admitted and maintained until the test has been made. The valve and pipe connections should be well protected with good non-conducting material, but the body of the instrument and the condenser should be left uncovered.

With a separator calorimeter, to find the quality of the steam use the following rule:

Rule 5.—*Divide the weight of water condensed in the condenser by the sum of the weights of the water condensed in the condenser and the entrained water collected in the separator.*

Or, let W = weight of condensed water;
w = weight of water in separator;
Q = quality of steam.

Then,
$$Q = \frac{W}{W + w}.$$

EXAMPLE.—The initial reading of the scale on the separator was .04 pound and the final reading .28 pound. The scale on the condenser indicated .6 pound at the beginning and 16.8 pounds at the ending of the test. What is the quality of the steam?

SOLUTION.—Water in separator = .28 − .04 = .24 pound. Water in condenser = 16.8 − .6 = 16.2 pounds. Applying rule **5**, we get

$$Q = \frac{16.2}{16.2 + .24} = .9854, \text{ or } 98.54 \text{ per cent. Ans.}$$

32. The quality of the steam having been determined, the actual amount of water evaporated by a boiler is found by multiplying the observed amount by the quality of the steam expressed decimally.

ANALYSIS OF COAL.

ULTIMATE ANALYSIS.

33. There are two analyses of coal that may be made, each of which furnishes information of considerable interest and value to the engineer. Of these analyses, one, called an **ultimate analysis,** determines the percentages of the various chemical elements of which the coal is composed, but does not necessarily show in what manner these elements are combined. It shows that, if a sample of the coal is separated into its elements, there will be certain proportions of oxygen, hydrogen, carbon, etc. These proportions are generally expressed as percentages of the weight of the original sample, the weight of which is considered as a unit, or 100 per cent. From the ultimate analysis, the heating value of the coal may easily be estimated by

Rule 6.—*To find the approximate heat of combustion of a pound of coal, multiply the weight of the carbon in hundredths of a pound by 14,600. Divide the weight of oxygen in hundredths of a pound by 8 and subtract the quotient from the weight of the hydrogen, also expressed in hundredths of a pound. Multiply the remainder by 62,000 and add the product to the first product. Multiply the weight of sulphur in hundredths of a pound by 4,000 and add the product to the previous sum.*

Or, let $X =$ heat of combustion of coal per pound;
$C =$ percentage of carbon;
$O =$ percentage of oxygen;
$H =$ percentage of hydrogen;
$S =$ percentage of sulphur.

Then, $X = 14,600\,C + 62,000\left(H - \frac{O}{8}\right) + 4,000\,S.$

EXAMPLE.—A coal contains 85 per cent. carbon, 4 per cent. oxygen, 6 per cent. hydrogen, 1 per cent. sulphur, and 4 per cent. ash. What is the heat of combustion per pound?

SOLUTION.—Applying rule **6**, we have

$$X = 14,600 \times .85 + 62,000\left(.06 - \frac{.04}{8}\right) + 4,000 \times .01 = 15,860 \text{ B. T. U.}$$ Ans.

The ultimate analysis of a fuel is a difficult and expensive operation that can be successfully performed only by a skilled chemist with the facilities of a well-appointed laboratory. Owing to its expense, it is seldom made except in such cases as important boiler trials, where it is desired to obtain the most accurate information possible regarding the properties of the fuel used.

PROXIMATE ANALYSIS.

34. Introduction.—Although the ultimate analysis of a fuel presents difficulties that render it impracticable for any but a skilled chemist, there is a method by means of which a careful engineer can acquire an amount of skill that will enable him to determine the percentages of water, volatile matter, fixed carbon, and ash with a fair degree of accuracy. This method is called a **proximate analysis.**

The instruments and devices necessarily required are a sensitive balance with a box of weights, a platinum or porcelain crucible and stand for it, a Bunsen burner, and a blast burner.

35. Selecting the Sample.—The first operation in making a fuel analysis consists in the selection of a sample that will fairly represent an average value of the fuel. Pieces of coal are selected at random here and there, until about 100 pounds have been collected. These should be broken up into pieces approximately equal in size and the pieces then thoroughly mixed. A conical pile is now heaped up, and by continually raking down a little coal from the

pile, a flat circular pile about 4 inches thick is formed, as shown in Fig. 4. Next mark off the pile into quadrants and remove two of the sectors diagonally opposite each other, as *1* and *3*, or *4* and *2*. The remaining coal should now be broken up into a smaller size, and the whole operation gone through again. This should be repeated until the coal has been pulverized and only about a pint measure full is left. This amount is then finely pulverized and spread out on a piece of glazed paper, in a circular form. It is now divided into, say, 8 equal sectors, and the amount to be subjected to analysis is then made up by taking equal parts from each sector in succession. Owing to the fact that the composition of the coal is to be expressed in per cent. of the weight of the sample, it is customary to use the metric system of weights, which is, in fact, the system universally used by chemists.

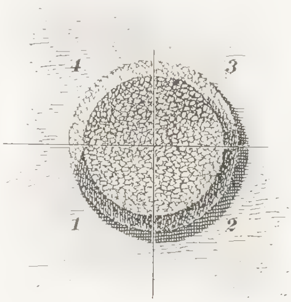


FIG. 4.

36. Analyzing the Sample.—The analysis of coal is performed in the following order: determination of moisture, determination of volatile matter, determination of carbon, and, finally, determination of the ash. It is based on the fact that the moisture and volatile matter can be driven off by heat and that by increasing the heat to the igniting temperature of carbon, the latter can be burned, leaving behind the incombustible substances known as *ash*.

In order that the results obtained may be compared with those obtained by others, it is essential that a uniform method be employed by all; that is, all chemists should use the same weight of coal to be analyzed and should heat it for the same length of time, as given in the directions for the standard laboratory method explained below.

37. To determine the moisture, weigh out exactly 1 gram of the sample into a porcelain or platinum crucible and put

in a drying oven in which a temperature of 225° F. is maintained. Keep it there for exactly 1 hour; allow it to cool, and then weigh the sample. The loss in weight represents the moisture. Thus, if the sample weighed 1 gram (1,000 milligrams) before heating and 912 milligrams after heating, the moisture weighed 1,000 − 912 = 88 milligrams. Reducing it to per cent., we have moisture $= \frac{88 \times 100}{1{,}000} = 8.8$ per cent.

38. The volatile matter is determined by weighing out 1.5 grams (1,500 milligrams) of the undried pulverized sample into a platinum crucible and covering tightly. Heat it for $3\frac{1}{2}$ minutes over a Bunsen burner, keeping up a bright red heat, and then heat it immediately, without cooling, over a blast burner for $3\frac{1}{2}$ minutes at a white heat. Allow the sample to cool and then weigh it. The loss in weight represents the volatile matter, i. e., the hydrocarbons and the moisture. Thus, if the sample weighed 1,500 milligrams before heating and 1,149 milligrams after heating, the volatile matter weighs 1,500 − 1,149 = 351 milligrams and constitutes $\frac{351 \times 100}{1{,}500} = 23.4$ per cent. of the coal. Then, since the moisture is 8.8 per cent., the volatile combustible is 23.4 − 8.8 = 14.6 per cent.

If a coke is formed in the preceding operation, make a note of its properties, color, firmness, etc.

39. To determine the fixed carbon, place what is left of the sample after having determined the volatile matter (1,149 milligrams in this case) into a crucible and weigh it. Place this in an inclined position, with cover removed, over a Bunsen burner and heat it until the carbon is all burned. By inclining the crucible and having it uncovered, the air can circulate freely in the crucible; this is essential, since the carbon cannot burn without the oxygen supplied by the air.

To determine positively when the carbon is all burned, heat until the ash appears to be free from carbon; then cool

the crucible and weigh. Place it over the lamp again and burn for a few minutes longer; then cool and weigh. If the second weight is the same as the first, the carbon is all burned out and the crucible contains nothing but ash. If, however, the second weight is less than the first, more carbon has been burned out, and the process should be repeated until two successive weighings give the same result. The combustion of the carbon may be hastened by stirring the charge occasionally with a platinum wire. The difference between the weight of the crucible with coke and its weight with the ash remaining after the carbon has been burned out is the fixed carbon.

Assume that the crucible and coke weighed 21,500 milligrams and after burning out the carbon it weighs 20,650 milligrams. Then the carbon weighs $21,500 - 20,650 = 850$ milligrams. Remembering that the original weight of the sample was 1,500 milligrams, the percentage of carbon is $\frac{850 \times 100}{1,500} = 56.67$ per cent.

40. The ash can either be weighed directly or its weight can be obtained by subtracting the weight of the crucible from the weight of the crucible and ash, i. e., 20,650 milligrams. Let the crucible weigh 20,351 milligrams. The ash then weighs $20,650 - 20,351 = 299$ milligrams. Remembering that the sample weighed 1,500 milligrams originally, the percentage of ash is $\frac{299 \times 100}{1,500} = 19.93$ per cent.

Summing up, the sample shows the coal to have the following composition:

Moisture	8.80 per cent.
Hydrocarbons	14.60 per cent.
Fixed carbon	56.67 per cent.
Ash	19.93 per cent.
	100.00 per cent.

41. In making the proximate analysis, great care must be taken to weigh as exactly as possible and also to avoid

spilling any part of the sample. The composition of the hydrocarbons not being given by the proximate analysis, the heating value of the coal cannot be accurately computed from the results obtained by it; the average composition of the volatile combustibles, however, varies in most coals but little from the composition of marsh gas, CH_4; it will, therefore, be found that a very good approximate estimate of the heating value can be made by calculating from the percentages of fixed carbon and volatile hydrocarbons determined by the proximate analysis, under the assumption that all the volatile matter is composed of CH_4.

Under this assumption, the rule for calculating the approximate heating value from the percentages given by the proximate analysis is as follows:

Rule 7.—*Multiply the percentage of hydrocarbons, expressed as a decimal, by 23,600, and the percentage of fixed carbon, expressed in the same way, by 14,600; the sum of these two products will be the approximate heating value of 1 pound of the undried coal.*

Or, $$X = 23{,}600\, V + 14{,}600\, C,$$

where V = percentage of hydrocarbons and the other letters have the same meaning as in rule **6**.

Example.—What is the heat of combustion of a coal having the composition given in Art. **40**?

Solution.—Applying rule **7**, we get

$$X = 23{,}600 \times .146 + 14{,}600 \times .5667 = 11{,}719.4 \text{ B. T. U.} \quad \text{Ans.}$$

STANDARD FORM OF BOILER TRIAL.

42. The American Society of Mechanical Engineers in 1885 accepted the report of a committee that had formulated a set of rules for the conducting of boiler trials in order to have uniformity. The code of rules was revised in 1899 by another committee and the universal adoption of the revised rules recommended. The rules are given below practically in full.

RULES FOR CONDUCTING BOILER TRIALS.

(*Code of 1899.*)

I. *Determine at the outset* the specific object of the proposed trial, whether it be to ascertain the capacity of the boiler, its efficiency as a steam generator, its efficiency and its defects under usual working conditions, the economy of some particular kind of fuel, or the effect of changes of design, proportion, or operation; and prepare for the trial accordingly.

II. *Examine the boiler*, both outside and inside; ascertain the dimension of grates, heating surfaces, and all important parts; and make a full record, describing the same, and illustrating special features by sketches. The area of heating surface is to be computed from the surfaces of shells, tubes, furnaces, and fireboxes in contact with the fire or hot gases. The outside diameter of water tubes and the inside diameter of fire tubes are to be used in the computation. All surfaces below the mean water level which have water on one side and products of combustion on the other are to be considered as water-heating surface, and all surfaces above the mean water level which have steam on one side and products of combustion on the other are to be considered as superheating surface.

III. *Notice the general condition* of the boiler and its equipment and record such facts in relation thereto as bear upon the objects in view.

If the object of the trial is to ascertain the maximum economy or capacity of the boiler as a steam generator, the boiler and all its appurtenances should be put in first-class condition. Clean the heating surface inside and outside, remove clinkers from the grates and from the sides of the furnace. Remove all dust, soot, and ashes from the chambers, smoke connections, and flues. Close air leaks in the masonry and poorly fitted cleaning doors. See that the damper will open wide and close tight. Test for air leaks by firing a few shovels of smoky fuel and immediately closing the damper,

observing the escape of smoke through the crevices, or by passing the flame of a candle over cracks in the brickwork.

IV. *Determine the character of the coal* to be used. For tests of the efficiency or capacity of the boiler for comparison with other boilers, the coal should, if possible, be of some kind which is commercially regarded as a standard. For New England and that portion of the country east of the Alleghany Mountains, good anthracite egg coal, containing not over 10 per cent. of ash, and semi-bituminous Clearfield (Pennsylvania), Cumberland (Maryland), and Pocahontas (Virginia) coals are thus regarded. West of the Alleghany Mountains, Pocahontas (Virginia) and New River (West Virginia) semi-bituminous, and Youghiogheny or Pittsburg bituminous coals are recognized as standards. There is no special grade of coal mined in the Western States which is widely recognized as of superior quality or considered as a standard coal for boiler testing. Big Muddy lump, an Illinois coal mined in Jackson County, Illinois, is suggested as being of sufficiently high grade to answer these requirements in districts where it is more conveniently obtainable than the other coals mentioned above.

For tests made to determine the performance of a boiler with a particular kind of coal, such as may be specified in a contract for the sale of a boiler, the coal used should not be higher in ash and in moisture than that specified, since increase in ash and moisture above the specified amount is apt to cause a falling off of both capacity and economy in greater proportion than the proportion of such increase.

V. *Establish the correctness of all apparatus* used in the test for weighing and measuring. These are:

1. Scales used for weighing coal, ashes, and water.

2. Tanks or water meters for measuring water. Water meters, as a rule, should only be used as a check on other measurements. For accurate work, the water should be weighed or measured in a tank.

3. Thermometers and pyrometers for taking temperatures of air, steam, feedwater, waste gases, etc.

4. Pressure gauges, draft gauges, etc.

The kind and location of the various pieces of testing apparatus must be left to the judgment of the person conducting the test, always keeping in mind the main object, i. e., to obtain authentic data.

VI. *See that the boiler is thoroughly heated* before the trial to its usual working temperature. If the boiler is new and of a form provided with a brick setting, it should be in regular use at least a week before the trial, so as to dry and heat the walls. If it has been laid off and become cold, it should be worked before the trial until the walls are well heated.

VII. The *boiler and connections* should be proved to be free from leaks before beginning a test, and all water connections, including blow and extra feedpipes, should be disconnected, stopped with blank flanges, or bled through special openings beyond the valves, except the particular pipe through which water is to be fed to the boiler during the trial. During the test, the blow-off and feedpipes should remain exposed to view. If an injector is used, it should receive steam directly through a felted pipe from the boiler being tested.

Note.—In feeding a boiler undergoing test with an injector taking steam from another boiler or from the main steam pipe from several boilers, the evaporative results may be modified by a difference in the quality of the steam from such source compared with that supplied by the boiler being tested, and in some cases the connection to the injector may act as a drip for the main steam pipe. If it is known that the steam from the main steam pipe is of the same pressure and quality as that furnished by the boiler undergoing the test, the steam may be taken from such main steam pipe.

If the water is metered after it passes the injector, its temperature should be taken at the point where it leaves the injector. If the quantity is determined before it goes to the injector, the temperature should be determined on the suction side of the injector; and if no change of temperature occurs other than that due to the injector, the temperature thus determined is properly that of the feedwater. When the temperature changes between the injector

and the boiler, as by the use of a heater or by radiation, the temperature at which the water enters and leaves the injector and that at which it enters the boiler should all be taken. In that case the weight to be used is that of the water leaving the injector, computed from the heat units if not directly measured, and the temperature that of the water entering the boiler.

Let w = weight of water entering the injector;
x = weight of steam entering the injector;
h_1 = heat units per pound of water entering injector;
h_2 = heat units per pound of steam entering injector;
h_3 = heat units per pound of water leaving injector.

Then, $w + x$ = weight of water leaving injector;

$$x = w \frac{h_3 - h_1}{h_2 - h_3}.$$

See that the steam main is so arranged that the water of condensation cannot run back into the boiler.

VIII. *Duration of the Test.*—For tests made to ascertain either the maximum economy or the maximum capacity of the boiler, irrespective of the particular class of service for which it is regularly used, the duration should be at least 10 hours of continuous running.

If the rate of combustion exceeds 25 pounds of coal per square foot of grate surface per hour, it may be stopped when a total of 250 pounds of coal has been burned per square foot of grate.

In cases where the service requires continuous running for the whole 24 hours of the day, with shifts of firemen a number of times during that period, it is well to continue the test for at least 24 hours.

When it is desired to ascertain the performance under the working conditions of practical running, whether the boiler be regularly in use 24 hours a day or only a certain number of hours out of each 24, the fires being banked the balance of the time, the duration should not be less than 24 hours.

IX. *Starting and Stopping a Test.*—The conditions of the boiler and furnace in all respects should be, as nearly as

possible, the same at the end as at the beginning of the test. The steam pressure should be the same; the water level the same; the fire upon the grates should be the same in quantity and condition; and the walls, flues, etc. should be of the same temperature. Two methods of obtaining the desired equality of conditions of the fire may be used, viz.: "the standard method" and "the alternate method," the latter being employed where it is inconvenient to make use of the standard method.

X. *Standard Method of Starting and Stopping a Test.*—Steam being raised to the working pressure, remove rapidly all the fire from the grate, close the damper, clean the ash-pit, and as quickly as possible start a new fire with weighed wood and coal, noting the time and the water level while the water is in a quiescent state, just before lighting the fire.

NOTE.—The gauge glass should not be blown out within an hour before the water level is taken at the beginning and end of a test, otherwise an error in the reading of the water level may be caused by a change in the temperature and density of the water in the pipe leading from the bottom of the glass into the boiler.

At the end of the test remove the whole fire, which has been burned low, clean the grates and ash-pit, and note the water level when the water is in a quiescent state, and record the time of hauling the fire. The water level should be as nearly as possible the same as at the beginning of the test. If it is not the same, a correction should be made by computation and not by operating the pump after the test is completed.

XI. *Alternate Method of Starting and Stopping a Test.*—The boiler being thoroughly heated by a preliminary run, the fires are to be burned low and well cleaned. Note the amount of coal left on the grate as nearly as it can be estimated; note the pressure of steam and the water level. Note the time and record it as the starting time. Fresh coal which has been weighed should be now fired. The ash-pits should be thoroughly cleaned at once after starting. Before the end of the test the fires should be burned low, just as before the start, and the fires cleaned in such a

manner as to leave a bed of coal on the grates of the same depth and in the same condition as at the start. When this stage is reached, note the time and record it as the stopping time. The water level and steam pressure should previously be brought as nearly as possible to the same point as at the start. If the water level is not the same as at the start, a correction should be made by computation and not by operating the pump after the test is completed.

XII. *Uniformity of Conditions.*—In all trials made to ascertain maximum economy or capacity, the conditions should be maintained uniformly constant. Arrangements should be made to dispose of the steam so that the rate of evaporation may be kept the same from beginning to end. This may be accomplished in a single boiler by carrying the steam through a waste steam pipe, the discharge from which can be regulated as desired. In a battery of boilers, in which only one is tested, the draft may be regulated on the remaining boilers, leaving the test boiler to work under a constant rate of production.

Uniformity of conditions should prevail as to the pressure of steam, the height of water, the rate of evaporation, the thickness of fire, the times of firing and quantity of coal fired at one time, and as to the intervals between the times of cleaning the fires.

The method of firing to be carried on in such tests should be dictated by the expert or person in responsible charge of the test, and the method adopted should be adhered to by the fireman throughout the test.

XIII. *Keeping the Records.*—Take note of every event connected with the progress of the trial, however unimportant it may appear. Record the time of every occurrence and the time of taking every weight and every observation.

The coal should be weighed and delivered to the fireman in equal proportions, each sufficient for not more than one hour's run, and a fresh portion should not be delivered until the previous one has all been fired. The time required to consume each portion should be noted, the time being

recorded at the instant of firing the last of each portion. It is desirable that at the same time the amount of water fed into the boiler should be accurately noted and recorded, including the height of the water in the boiler and the average pressure of steam and temperature of feed during the time. By thus recording the amount of water evaporated by successive portions of coal, the test may be divided into several periods, if desired, and the degree of uniformity of combustion, evaporation, and economy analyzed for each period. In addition to these records of the coal and the feedwater, half hourly observations should be made of the temperature of the feedwater, of the flue gases, of the external air in the boiler room, of the temperature of the furnace when a furnace pyrometer is used, also of the pressure of steam, and of the readings of the instruments for determining the moisture in the steam. A log should be kept on properly prepared blanks containing columns for records of the various observations.

When the "standard method" of starting and stopping the test is used, the hourly rate of combustion and of evaporation and the horsepower should be computed from the records taken during the time when the fires are in active condition. This time is somewhat less than the actual time which elapses between the beginning and end of the run. The loss of time due to kindling the fire at the beginning and burning it out at the end makes this course necessary.

XIV. *Quality of Steam.*—The percentage of moisture in the steam should be determined by the use of either a throttling or a separating steam calorimeter. The sampling nozzle should be placed in the vertical steam pipe rising from the boiler. It should be made of $\frac{1}{2}$-inch pipe and should extend across the diameter of the steam pipe to within $\frac{1}{2}$ inch of the opposite side, being closed at the end and perforated with not less than twenty $\frac{1}{8}$-inch holes equally distributed along and around its cylindrical surface, but none of these holes should be nearer than $\frac{1}{2}$ inch to the inner side of the steam pipe. The calorimeter and the pipe leading to it should be well covered with felting. Whenever

the indications of the throttling or separating calorimeter show that the percentage of moisture is irregular or occasionally in excess of 3 per cent., the results should be checked by a steam separator placed in the steam pipe as close to the boiler as convenient, with a calorimeter in the steam pipe just beyond the outlet from the separator. The drip from the separator should be caught and weighed and the percentage of moisture computed therefrom added to that shown by the calorimeter.

Superheating should be determined by means of a thermometer placed in a mercury well inserted in the steam pipe. The degree of superheating should be taken as the difference between the reading of the thermometer for superheated steam and the readings of the same thermometer for saturated steam at the same pressure as determined by a special experiment and not by reference to Steam Tables.

XV. *Sampling the Coal and Determining Its Moisture.*—As each barrow load or fresh portion of coal is taken from the coal pile, a representative shovelful is selected from it and placed in a barrel or box in a cool place and kept until the end of the trial. The samples are then mixed and broken into pieces not exceeding 1 inch in diameter and reduced by the process of repeated quartering and crushing until a final sample weighing about 5 pounds is obtained and the size of the larger pieces is such that they will pass through a sieve with $\frac{1}{4}$-inch meshes. From this sample 2 one-quart, air-tight, glass preserving jars, or other air-tight vessels which will prevent the escape of moisture from the sample, are to be promptly filled, and these samples are to be kept for subsequent determinations of moisture and of heating value and for chemical analyses. During the process of quartering, when the sample has been reduced to about 100 pounds, a quarter to a half of it may be taken for an approximate determination of moisture. This may be made by placing it in a shallow iron pan, not over 3 inches deep, carefully weighing it, and setting the pan in the hottest place that can be found on the brickwork of the boiler setting or flues, keeping it there for at least 12 hours, and

then weighing it. The determination of moisture thus made is believed to be approximately accurate for anthracite and semi-bituminous coals, and also for Pittsburg or Youghiogheny coal; but it cannot be relied upon for coals mined west of Pittsburg or for other coals containing inherent moisture. For these latter coals it is important that a more accurate method be adopted. The method recommended by the committee for all accurate tests, whatever the character of the coal, is described as follows:

Take one of the samples contained in the glass jars and subject it to a thorough air-drying by spreading it in a thin layer and exposing it for several hours to the atmosphere of a warm room, weighing it before and after, thereby determining the quantity of surface moisture it contains. Then crush the whole of it by running it through an ordinary coffee mill adjusted so as to produce somewhat coarse grains (less than $\frac{1}{16}$ inch), thoroughly mix the crushed sample, select from it a portion of from 10 to 50 grams, weigh it in a balance which will easily show a variation as small as 1 part in 1,000, and dry it in an air or sand bath at a temperature between 240 and 280° F. for 1 hour. Weigh it and record the loss, then heat and weigh it again repeatedly at intervals for an hour or less, until the minimum weight has been reached and the weight begins to increase by oxidation of a portion of the coal. The difference between the original and the minimum weight is taken as the moisture in the air-dried coal. This moisture test should preferably be made on duplicate samples, and the results should agree within .3 to .4 of 1 per cent., the mean of the two determinations being taken as the correct result. The sum of the percentage of moisture thus found and the percentage of surface moisture previously determined is the total moisture.

XVI. *Treatment of Ashes and Refuse.*—The ashes and refuse are to be weighed in a dry state. If it is found desirable to show the principal characteristics of the ash, a sample should be subjected to a proximate analysis and the actual amount of incombustible material determined. For elaborate

trials a complete analysis of the ash and refuse should be made.

XVII. *Calorific Tests and Analysis of Coal.*—The quality of the fuel should be determined either by heat test or by analysis, or by both.

The rational method of determining the total heat of combustion is to burn the sample of the coal in an atmosphere of oxygen gas, the coal to be sampled as directed in Article XV. of this code.

The chemical analysis of the coal should be made only by an expert chemist. The total heat of combustion computed from the results of the ultimate analysis may be obtained by the use of Dulong's formula (with constants modified by recent determinations), viz.: $14,600\ C + 62,000 \left(H - \frac{O}{8}\right) + 4,000\ S$, in which C, H, O, and S refer to the proportions of carbon, hydrogen, oxygen, and sulphur, respectively, as determined by the ultimate analysis.

It is desirable that a proximate analysis should be made, thereby determining the relative proportions of volatile matter and fixed carbon. These proportions furnish an indication of the leading characteristics of the fuel and serve to fix the class to which it belongs. As an additional indication of the characteristics of the fuel, the specific gravity should be determined.

XVIII. *Analysis of Flue Gases.*—The analysis of the flue gases is an especially valuable method of determining the relative value of different methods of firing or of different kinds of furnaces. In making these analyses great care should be taken to procure average samples, since the composition is apt to vary at different points of the flue. The composition is also apt to vary from minute to minute, and for this reason the drawings of gas should last a considerable period of time. Where complete determinations are desired, the analyses should be intrusted to an expert chemist.

For the continuous indication of the amount of carbonic acid present in the flue gases, an instrument may be

employed which shows the weight of the sample of gas passing through it.

XIX. *Smoke Observations.*—It is desirable to have a uniform system of determining and recording the quantity of smoke produced where bituminous coal is used. The system commonly employed is to express the degree of smokiness by means of percentages dependent upon the judgment of the observer. The committee does not place much value upon the percentage method, because it depends so largely upon the personal element, but if this method is used, it is desirable that so far as possible a definition be given in explicit terms as to the basis and method employed in arriving at the percentage. The actual measurement of a sample of soot and smoke by some form of meter is to be preferred.

XX. *Miscellaneous.*—In tests for purposes of scientific research, in which the determination of all the variables entering into the test is desired, certain observations should be made which are, in general, unnecessary for ordinary tests. These are the measurement of the air supply, the determination of its contained moisture, the determination of the amount of heat lost by radiation, of the amount of infiltration of air through the setting, and (by condensation of all the steam made by the boiler) of the total heat imparted to the water.

As these determinations are rarely undertaken, it is not deemed advisable to give directions for making them.

XXI. *Calculations of Efficiency.*—Two methods of defining and calculating the efficiency of a boiler are recommended. They are:

1. Efficiency of the boiler

$$= \frac{\text{Heat absorbed per pound combustible}}{\text{Calorific value of 1 pound combustible}}.$$

2. Efficiency of the boiler and grate

$$= \frac{\text{Heat absorbed per pound coal}}{\text{Calorific value of 1 pound coal}}.$$

The first of these is sometimes called the efficiency based on combustible and the second the efficiency based on coal. The first is recommended as a standard of comparison for all tests, and this is the one which is understood to be referred to when the word "efficiency" alone is used without qualification. The second, however, should be included in a report of a test, together with the first, whenever the object of the test is to determine the efficiency of the boiler and furnace together with the grate (or mechanical stoker), or to compare different furnaces, grates, fuels, or methods of firing. The heat absorbed per pound of combustible (or per pound of coal) is to be calculated by multiplying the equivalent evaporation from and at 212° per pound combustible (or coal) by 965.7.

XXII. *The Heat Balance.*—An approximate "heat balance," or statement of the distribution of the heating value of the coal among the several items of heat utilized and heat lost, may be included in the report of a test when analyses of the fuel and of the chimney gases have been made. It should be reported in the form shown on the following page.

XXIII. *Report of the Trial.*—The data and results should be reported in the manner given in either one of the two following tables, omitting lines where the tests have not been made as elaborately as provided for in such tables. Additional lines may be added for data relating to the specific object of the test. The extra lines should be classified under the headings provided in the tables and numbered as per preceding line, with sub letters *a*, *b*, etc. The Short Form of Report, Table No. 2, is recommended for commercial tests and as a convenient form of abridging the longer form for publication when saving of space is desirable. For elaborate trials, it is recommended that the full log of the trial be shown graphically by means of a chart.

In Table No. 1 the items printed in italics correspond to the items in Table No. 2.

HEAT BALANCE, OR DISTRIBUTION OF THE HEATING VALUE OF THE COMBUSTIBLE.

Total Heat Value of 1 pound of Combustible. . B. T. U.

	B. T. U.	Per Cent.
1. Heat absorbed by the boiler = evaporation from and at 212° per pound of combustible × 965.7		
2. Loss due to moisture in coal = per cent. of moisture referred to combustible ÷ 100 × $(212 - t) + 966 + .48\,(T - 212)$ (t = temperature of air in the boiler room, T = that of the flue gases)		
3. Loss due to moisture formed by the burning of hydrogen = per cent. of hydrogen to combustible ÷ 100 × 9 × $[(212 - t) + 966 + 0.48\,(T - 212)]$		
4. Loss due to heat carried away in the dry chimney gases = weight of gas per pound of combustible × .24 × $(T - t)$		
5. Loss due to incomplete combustion of carbon = $\frac{CO}{CO_2 + CO} \times \frac{\text{per cent. } C \text{ in combustible}}{100} \times 10{,}150.$		
6. Loss due to unconsumed hydrogen and hydrocarbons, to heating the moisture in the air, to radiation, and unaccounted for. (Some of these losses may be separately itemized if data are obtained from which they may be calculated.)		
Totals.	. . . .	100.00

TABLE NO. 1.

DATA AND RESULTS OF EVAPORATIVE TEST.

ARRANGED IN ACCORDANCE WITH THE COMPLETE FORM ADVISED BY THE BOILER TEST COMMITTEE OF THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS. CODE OF 1899.

Made by .... on .... boiler, at ....
to determine ....
....

Principal conditions governing the trial ....
....
....
....

Kind of Fuel ....
Kind of Furnace ....
State of the weather ....
Method of starting and stopping the test ("standard" or "alternate," Arts. X and XI, Code.) ....

1. *Date of Trial* ....
2. *Duration of Trial* ....hours

DIMENSIONS AND PROPORTIONS.

A complete description of the boiler and drawings of the same, if of unusual type, should be given on an annexed sheet.

3. *Grate surface* .... *width* ....
   length .... *area* ....square feet
4. Height of furnace ....inches
5. Approximate width of air spaces in grate ....inches
6. Proportion of air space to whole grate surface . . per cent.
7. *Water-heating surface* ....square feet
8. *Superheating surface* ....square feet
9. Ratio of water-heating surface to grate surface . . — to 1
10. Ratio of minimum draft area to grate surface . . . 1 to —

Average Pressure.

11. *Steam pressure by gauge*.......pounds per square inch
12. *Force of draft between damper and boiler*.inches of water
13. Force of draft in furnace..............inches of water
14. Force of draft or blast in ash-pit........inches of water

Average Temperature.

15. Of external air..............................degrees
16. Of fireroom..................................degrees
17. Of steam.....................................degrees
18. Of feedwater entering heater.................degrees
19. Of feedwater entering economizer............degrees
20. *Of feedwater entering boiler*.................degrees
21. *Of escaping gases from boiler*...............degrees
22. Of escaping gases from economizer...........degrees

Fuel.

23. Size and condition.................................
24. Weight of wood used in lighting fire...........pounds
25. *Weight of coal as fired**......................pounds
26. *Percentage of moisture in coal*...............per cent.
27. *Total weight of dry coal consumed*............pounds
28. *Total ash and refuse*..........................pounds
29. Quality of ash and refuse.........................
30. Total combustible consumed.................pounds
31. *Percentage of ash and refuse in dry coal*......per cent.

Proximate Analysis of Coal.

		Of Coal.	Of Combustible.
32.	Fixed carbon.............	per cent.	per cent.
33.	Volatile matter...........	per cent.	per cent.
34.	Moisture..................	per cent.	per cent.
35.	Ash......................	per cent.	per cent.
		100 per cent.	100 per cent.
36.	Sulphur, separately determined..................	per cent.	per cent.

* Including Item 24 multiplied by 0.4.

Ultimate Analysis of Dry Coal.
(Art. XVII., Code.)

		Of Coal.	Of Combustible.
37.	Carbon (C)..............	per cent.	per cent.
38.	Hydrogen (H)...........	per cent.	per cent.
39.	Oxygen (O)..............	per cent.	per cent.
40.	Nitrogen (N)............	per cent.	per cent.
41.	Sulphur (S)..............	per cent.	per cent.
42.	Ash.....................	per cent.	per cent.
		100 per cent.	100 per cent.
43.	Moisture in sample of coal as received.............	per cent.	per cent.

Analysis of Ash and Refuse.

44. Carbon..................................per cent.
45. Earthy matter..............................per cent.

Fuel per Hour.

46. *Dry coal consumed per hour*..................pounds
47. Combustible consumed per hour..............pounds
48. *Dry coal per square foot of grate surface per hour*....................................pounds
49. Combustible per square foot of water-heating surface per hour..........................pounds

Calorific Value of Fuel.
(Art. XVII., Code.)

50. *Calorific value by oxygen calorimeter, per pound of dry coal*.............................B. T. U.
51. *Calorific value by oxygen calorimeter, per pound of combustible*..........................B. T. U.
52. Calorific value by analysis, per pound of dry coal.....................................B. T. U.
53. Calorific value by analysis, per pound of combustible...............................B. T. U.

Quality of Steam.

54. *Percentage of moisture in steam*..............per cent.
55. *Number of degrees of superheating*...........degrees
56. Quality of steam (dry steam unity)..................

WATER.

57. *Total weight of water fed to boiler* ....pounds

58. Equivalent water fed to boiler from and at 212 ....pounds

59. *Water actually evaporated, corrected for quality of steam* ....pounds

60. Factor of evaporation* ....pounds

61. Equivalent water evaporated into dry steam from and at 212° (Item 59 times Item 60)....pounds

WATER PER HOUR.

62. *Water evaporated per hour, corrected for quality of steam*..............................pounds

63. *Equivalent evaporation per hour from and at 212°* ....pounds

64. *Equivalent evaporation per hour from and at 212° per square foot of water-heating surface*......pounds

HORSEPOWER.

65. *Horsepower developed.* (*$34\frac{1}{2}$ pounds of water evaporated per hour into dry steam from and at 212° equals 1 horsepower*).........horsepower

66. *Builders' rated horsepower* ....horsepower

67. *Percentage of builders' rated horsepower developed* ....per cent

ECONOMIC RESULTS.

68. *Water apparently evaporated under actual conditions per pound of coal as fired.* (*Item 57 divided by Item 25*)..........................pounds

69. *Equivalent evaporation from and at 212° per pound of coal as fired.* (*Item 61 divided by Item 25*)...................................pounds

* The factor of evaporation is to be computed here from the formula $f = \frac{H - t + 32}{965.7}$, where f = factor of evaporation, H = total heat of steam, and t = the feedwater temperature. The factor of evaporation thus calculated will differ slightly from that calculated by the formula in Art. **21,** on account of the committee having taken 965.7 B. T. U. as the latent heat of steam at 212° instead of the more widely accepted value of 966.1 B. T. U. The difference in results due to this is but very slight.

70. *Equivalent evaporation from and at 212° per pound of dry coal. (Item 61 divided by Item 29)*.........pounds

71. *Equivalent evaporation from and at 212° per pound of combustible. (Item 61 divided by Item 30)*.........pounds

(If the equivalent evaporation, Items 69, 70, and 71, is not corrected for the quality of the steam, the fact should be stated.)

EFFICIENCY.

(*Art. XXI., Code.*)

72. *Efficiency of the boiler; heat absorbed by the boiler per pound of combustible divided by the heat value of 1 pound of combustible*....per cent.

73. *Efficiency of boiler, including the grate; heat absorbed by the boiler, per pound of dry coal, divided by the heat value of 1 pound of dry coal*.........per cent.

COST OF EVAPORATION.

74. *Cost of coal per ton of......pounds delivered in boiler room*.........$

75. Cost of fuel for evaporating 1,000 pounds of water under observed conditions.........$

76. *Cost of fuel used for evaporating 1,000 pounds of water from and at 212°*.........$

SMOKE OBSERVATIONS.

77. Percentage of smoke as observed.........per cent.

78. Weight of soot per hour obtained from smoke meter.........ounces

79. Volume of soot per hour obtained from smoke meter.........cubic inches

METHODS OF FIRING.

80. Kinds of firing (spreading, alternate, or coking)......

81. Average thickness of fire.........

82. Average intervals between firings for each furnace during time when fires are in normal condition......
83. Average intervals between times of leveling or breaking up ....

Analyses of the Dry Gases.

84. Carbon dioxide (CO_2).................. per cent.
85. Oxygen (O).......................... per cent.
86. Carbon monoxide (CO)................. per cent.
87. Hydrogen and hydrocarbons............ per cent.
88. Nitrogen (by difference) (N) .... per cent.

100 per cent.

TABLE NO. 2.

DATA AND RESULTS OF EVAPORATIVE TEST.

Arranged in Accordance with the Short Form Advised by the Boiler Test Committee of the American Society of Mechanical Engineers. Code of 1899.

Made by............on............boiler, at.............
to determine ....
Kind of fuel...
Kind of furnace..
Method of starting and stopping the test ("standard" or "alternate," Arts. X and XI., Code) ....
Grate surface..............................square feet
Water-heating surface......................square feet
Superheating surface.......................square feet

Total Quantities.

1. Date of trial ....
2. Duration of trial..............................hours
3. Weight of coal as fired.........................pounds
4. Percentage of moisture in coal...............per cent.
5. Total weight of dry coal consumed............pounds
6. Total ash and refuse.........................pounds
7. Percentage of ash and refuse in dry coal......per cent.
8. Total weight of water fed to the boiler........pounds

9 Water actually evaporated, corrected for moisture or superheat in steam..................pounds

10 Equivalent water evaporated into dry steam from and at 212°.................. ....pounds

Hourly Quantities.

11. Dry coal consumed per hour.................pounds

12. Dry coal per square foot of grate surface per hour .. .. pounds

13 Water evaporated per hour, corrected for quality of steam................................pounds

14 Equivalent evaporation per hour from and at 212°....................................pounds

15. Equivalent evaporation per hour from and at 212° per square foot of water-heating surface ....pounds

Average Pressures, Temperatures, Etc.

16. Steam pressure by gauge......pounds per square inch

17. Temperature of feedwater entering boiler......degrees

18. Temperature of escaping gases from boiler.....degrees

19 Force of draft between damper and boiler ....inches of water

20. Percentage of moisture in steam, or number of degrees of superheating ....per cent. or degrees

Horsepower.

21 Horsepower developed (Item 14 divided by 34½) . ... horsepower

22. Builders' rated horsepower................horsepower

23 Percentage of builders' rated horsepower developed . .. .per cent.

Economic Results.

24. Water apparently evaporated under actual conditions per pound of coal as fired. (Item 8 divided by Item 3)........................pounds

25. Equivalent evaporation from and at 212° per pound of coal as fired. (Item 10 divided by Item 3) ....pounds
26. Equivalent evaporation from and at 212° per pound of dry coal. (Item 10 divided by Item 5)....................................pounds
27. Equivalent evaporation from and at 212° per pound of combustible. [Item 10 divided by (Item 5 minus Item 6)]....................pounds

(If Items 25, 26, and 27 are not corrected for quality of steam, the fact should be stated.)

EFFICIENCY.

28. Calorific value of dry coal per pound.........B. T. U.
29. Calorific value of the combustible per pound ....B. T. U.
30. Efficiency of boiler (based on combustible) ....per cent.
31. Efficiency of boiler, including grate (based on dry coal) ....per cent.

COST OF EVAPORATION.

32. Cost of coal per ton of .... pounds delivered in boiler room...............................$
33. Cost of coal required for evaporating 1,000 pounds of water from and at 212°....................$

WORKING UP THE DATA.

43. The method of working up the data obtained during a boiler trial is shown by the following example:

ILLUSTRATIVE EXAMPLE.—Given the following data observed during a boiler trial, required, to make the necessary calculations for economic evaporation and horsepower:

Duration of test.	10 hours.
Average gauge pressure.	72 pounds.
Average temperature of feedwater	122° F.
Pounds of coal burned	15,232 pounds.
Percentage of ash.	$4\frac{1}{2}$ per cent.
Water fed to boiler	124,600 pounds.
Average quality of steam	97.2 per cent.
Rated horsepower . . .	300

CALCULATIONS. — Water evaporated = 124,600 × .972 = 121,111.2 pounds.

Water evaporated per pound of coal—actual conditions = 121,111.2 ÷ 15,232 = 7.95 pounds.

Water evaporated per pound of combustible

$$= 121{,}111.2 \div \left[15{,}232 \times \frac{(100 - 4\frac{1}{2})}{100}\right] = 121{,}111.2 \div 14{,}546.6$$

= 8.33 pounds.

From the table, Art. **22**, the factor of evaporation for the given pressure and temperature of feedwater is 1.1264.

Hence, the equivalent evaporation from and at 212° per pound of coal is 7.95 × 1.1264 = 8.96 pounds.

The equivalent evaporation per pound of combustible is 8.33 × 1.1264 = 9.38 pounds.

The total equivalent evaporation from and at 212° F. per hour is

$$\frac{121{,}111.2 \times 1.1264}{10} = 13{,}641.966 \text{ pounds.}$$

The horsepower is, therefore,

$$13{,}641.966 \text{ lb.} \div 34\tfrac{1}{2} = 395.42 \text{ horsepower.}$$

The per cent. above rated capacity is

$$\frac{(395.42 - 300) \times 100}{300} = 31.81 \text{ per cent.}$$

BOILER FEEDING AND FEEDWATER PROBLEMS.

BOILER FEEDING.

APPARATUS USED FOR BOILER FEEDING.

1. A boiler may be fed by any one of three different classes of apparatus, and often two different classes of apparatus are used alongside of each other, one set of apparatus being held in reserve for emergencies. The three classes of apparatus used are the following: (1) *Pumps.* A boiler feed-pump may be a hand pump, which is used only for very small boilers, or it may be a direct-acting steam pump, or an attached pump driven directly from the engine, or a belt-driven pump, etc. Under special conditions an electrically driven pump may be used. Pumps will not be considered here. (2) *Injectors and inspirators.* These are steam-actuated devices. (3) *Gravity-feed apparatus.* In a device of this kind, the feedwater flows into the boiler by gravity.

INJECTORS.

2. Introduction.—The **injector** is an apparatus for forcing the feedwater into a steam boiler. It was invented in 1858 by an eminent French scientist, Henri Giffard, and was introduced into the United States in 1860 by William Sellers and Company, of Philadelphia.

3. On investigating the action of the injector, it will be found that dry steam at a given pressure enters the apparatus, passes through several contracted passages, raises several check-valves, and then forces water into the boiler against a pressure equal to that which the steam had when it first began the operation. The steam, in forcing the water through the injector and into the boiler, gives up its heat and performs actual mechanical work as truly as though the steam acted on a piston and moved a pump plunger with it.

4. Fundamental Principle of Action.—Before the action of an actual injector can be studied properly, it is necessary to have a clear understanding of the fundamental principle on which its action is based. This may be stated thus: *A current of any kind, be it steam, air, water, or other matter, by reason of friction has a tendency to induce a movement in the same direction of any body with which it may come in contact.*

The steam entering an injector and moving with an extremely high velocity first carries the air inside the injector with it and thus creates a partial vacuum, causing the water to flow into it. The steam then imparts a portion of its velocity to the water and gives it sufficient momentum to throw open the check-valves and enter the boiler. By striking the cold mass of water, the heat and velocity of the steam will be greatly reduced, and it will be condensed at the same time.

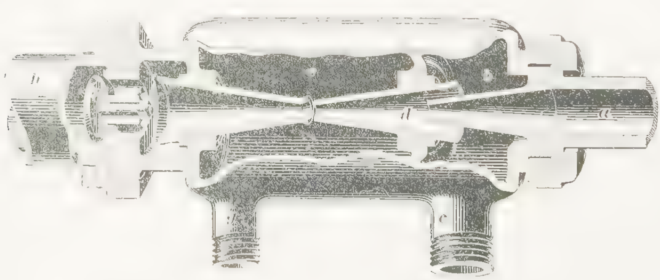


FIG. 1.

5. Essential Parts.—The essential parts of an injector are shown in Fig. 1, which does not represent an actual

injector, but serves to illustrate the combination of the essential parts by means of which the injector performs its function. Steam is admitted from the boiler through the **steam pipe** *a*; the chamber *b* connects to the water supply through the nozzle *c*. The tube *d* is called the **combining tube**; the space *e* is the **overflow space**; the overflow is carried away by a pipe attached to the **overflow nozzle** *g*. The water passes through the **check-valve** *i* into the discharge pipe *h* and thence into the boiler. The check-valve may not be part of the injector itself, but it is an essential part of the installation.

6. Action.—The action of the instrument is as follows: Steam is admitted through *a* and flows through the nozzle with a high velocity, passes through the combining tube *d* and out through the overflow *e* and nozzle *g*. This current of steam carries the air in the chamber *b* with it, thus forming a partial vacuum; the pressure of the atmosphere then forces water from the supply into the chamber and into the combining tube *d*. In *d* the steam and water are combined, with the result that the steam imparts a great deal of its velocity to the water and at the same time is condensed. This forms a jet of water that flows from the combining tube *d* with such a high velocity that it passes over the overflow *e* and into the discharge pipe *h*, the energy in the water being great enough to overcome the pressure in the boiler. The water thus flows past the check-valve *i* into the boiler.

7. Effect of Steam Supply.—When the injector is working properly, all the steam that is used to give the water its high velocity is condensed, thus leaving a steady, unbroken jet of water that flows across the space between the combining tube and the discharge pipe. If the water is too hot to condense the steam, or if there is so much steam that it is not all condensed in the combining tube, the steam, owing to its lightness, will not be carried into the boiler, but will flow out through *e* into the overflow space and be discharged from the overflow nozzle *g*. This escaping steam breaks the jet of water and interferes with the action of the

instrument so much as to stop the flow into the boiler, and serves to show the engineer when there is too much steam admitted for the water that is used. When the supply of steam is too small, the velocity of the jet of water flowing from the nozzle is so small that its momentum is not sufficient to carry it into the discharge pipe against the pressure in the boiler, and the water is therefore discharged through *c* and out of the overflow nozzle *g*. This shows the engineer that the supply of steam is too small.

8. Temperature of Feedwater Delivered by an Injector.—The temperature at which water will be delivered to the boiler depends on the steam pressure and on the quantity of water being delivered per pound of steam, the feedwater temperature remaining constant. Thus, if an injector is worked at its maximum capacity with a steam pressure of 30 pounds, the temperature of the water delivered to the boiler will be about 114° F.; if the pressure is 200 pounds, the temperature of the injected water will be about 154° F. The temperature of the water delivered will increase as the capacity of the injector is cut down from its maximum to its minimum. Thus, under 140 pounds steam pressure and working at its maximum capacity, an injector may deliver water at about 135° F.; while if the injector is cut down to its minimum capacity, the water would be delivered at about 250° F. Under ordinary working conditions the water is probably delivered at a temperature between 160° and 200° F.

The highest temperature at which an injector will lift the feedwater *decreases* as the steam pressure under which the injector is working is *increased*. At low steam pressures the injector may raise water at 125° or 130° F., while at 140 pounds and upwards it is not safe to have the water much above 110° F.

9. Effect of Steam Pressure on Water Delivered. The number of pounds of water delivered per pound of steam *decreases* as the steam pressure is *increased*. At 30 pounds steam pressure, an injector may deliver from

20 to 25 pounds of water per pound of steam, while at 140 pounds pressure it will deliver only about 13 pounds, and at 180 only about 11 pounds.

10. Range of an Injector.—The term **range,** when applied to an injector, refers to the steam pressure at which it will start and the steam pressure at which it ceases to work. The range of an injector decreases with any increase in the distance that the water must be lifted, and also decreases with any increase in the temperature of the water supply. This is clearly shown in the following table published by the American Injector Company and referring to an injector manufactured by it.

TABLE I.

RANGE OF INJECTORS.

Vertical Lift. Feet.	Feedwater at 60°.		Feedwater at 75°.		Feedwater at 100°.	
	Starting Pressure.	Stopping Pressure.	Starting Pressure.	Stopping Pressure.	Starting Pressure.	Stopping Pressure.
2	15	155	15	145	20	120
4	18	150	18	140		
6	20	142				
8	25	135	25	125		
10	30	125	30	115	35	90
12	35	118				
14	40	110				
15			50	85	45	70
16	45	102				
18	50	90				
20	55	85	55	75		
22	55	75				

11. The steam pressure at which injectors of different makes will start varies somewhat, but the range between

the starting and stopping pressures with different injectors is practically the same. Most injectors will start on 25 pounds steam pressure, but some are made to start on 15 pounds.

CONSTRUCTION OF INJECTORS.

12. Classification of Injectors.—Injectors may be divided into two general classes, the *non-lifting* and *lifting* injectors. They differ from each other, as implied by the name, in that the one class is capable of lifting the water from a level lower than its own, which the other class cannot do.

13. Non-lifting injectors are intended for use where there is a head of water available, consequently they must be placed below the water level of the supply tank, if one is used. When the water comes to a non-lifting injector under pressure, as from a city main, it can be placed in almost any convenient position close to the boiler. Non-lifting injectors resemble the lifting injector so much in their action that no description of them will be given

14. Lifting injectors are of two distinct types called **automatic** and **positive** injectors. Since positive injectors generally have two sets of tubes, they are frequently called **double-tube** injectors.

15. Automatic injectors are so called from the fact that they will automatically start again in case the jet of water is broken by jarring or other means. They are simpler in construction than double-tube injectors, and for a moderate temperature of feedwater supply and not too great a range in steam variation answer very well. They are very generally used on stationary and portable boilers and traction engines.

16. Positive, or double-tube, injectors are provided with two sets of tubes, one set of which is used for lifting the water, while the other set forces the water thus delivered to it into the boiler. A positive injector has a wider range

than an automatic injector and will handle a hotter feed-water supply. It will also lift water to a greater vertical height than the automatic injector.

17. Automatic Injectors.—The construction of the **Penberthy automatic injector** is shown in Fig. 2. Steam

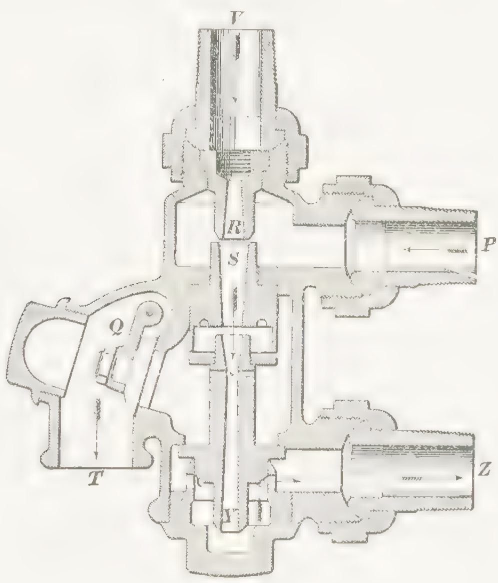


FIG. 2.

from the boiler enters the nipple V and passes into the nozzle R and then into the conical combining tube S. In rushing past the annular opening between R and S it creates a partial vacuum and causes water to flow through P, filling the space surrounding the lower end of R and upper end of S. The nipple P, which is shown at the right-hand side, is really situated in the rear. At first the mingled steam and water, by reason of the water not having acquired sufficient momentum, do not flow to the boiler; but after the tube Y, the space surrounding it, and the feed-delivery pipe attached to the nipple Z are filled, the mingled steam and water force the swing check-valve Q and pass through the

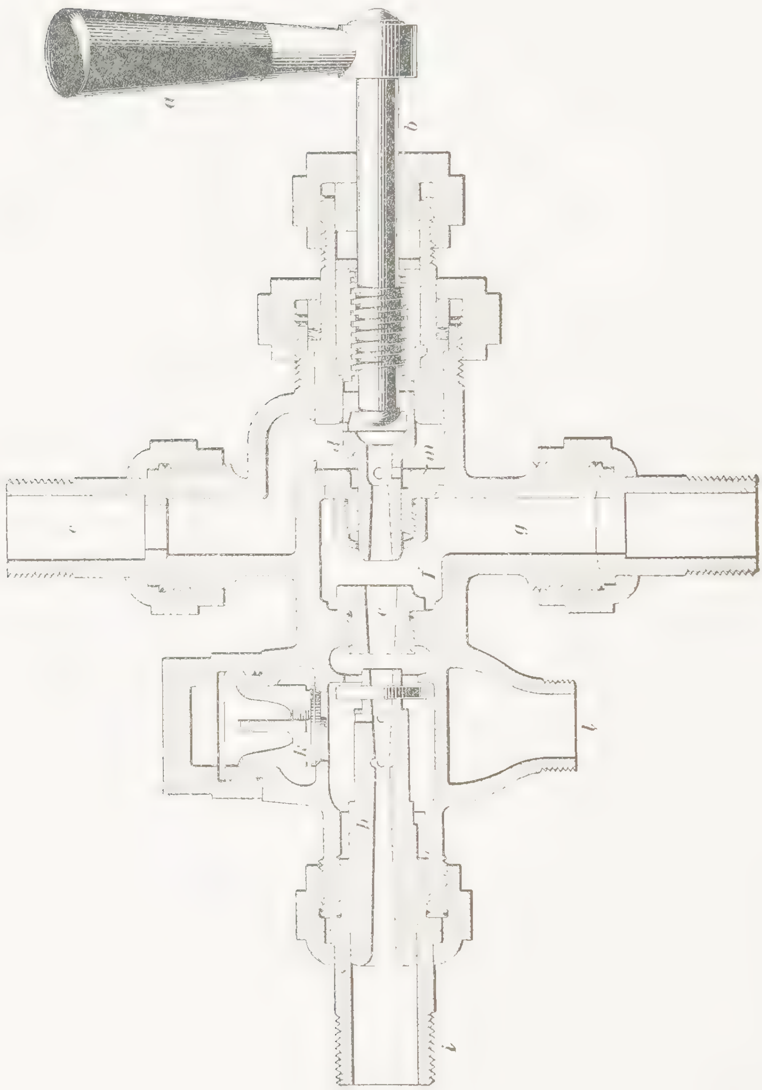


FIG. 3

overflow *T*. As soon as the jet of water passing through the combining tube has acquired sufficient momentum, the boiler check-valve is forced open and the water commences to enter the boiler. In consequence, no more water will enter the space around the lower end of *S* and the upper end of *Y*, and there being no pressure in this space, the overflow valve *Q* will close. The overflow valve is kept closed by the atmospheric pressure on top of it, for while the injector is working steadily, there will be a partial vacuum in the space around *S* and *Y*.

18. To start the injector, all that is required is to turn on the steam and water. If the steam supply is too great, steam will issue from the overflow; if the water supply is too great, water will issue. Should the jet of water be broken, i. e., fail to enter the boiler, the overflow valve will lift and the mingled water and steam will come out of the overflow until the jet has acquired sufficient momentum to enter the boiler again, when the overflow valve will close for the reasons given in the preceding article.

19. The automatically closing overflow valve is the distinguishing feature of the automatic injector, and in some form or other is found in all instruments of this class.

20. Fig. 3 is a sectional view of the **Buffalo automatic injector,** which differs considerably in its construction from the Penberthy, but which operates in practically the same manner. This injector needs no valves on the steam and water pipes, the steam admission valve being controlled by a handle *a* placed on the valve stem *b*. With the valve and stem in the position shown, the injector is working. Steam passes through *c* into a chamber surrounding the steam nozzle *d* and through openings in the rear end of the nozzle into the latter. In rushing into the suction jet *e*, it carries the air in *f* with it, creating a partial vacuum there and causing water to flow through *g*. This water, combined with the steam, enters the combining tube *h* and fills the boiler feedpipe, which is connected to the nipple *i*. At first the jet has not sufficient momentum to force the boiler

check-valve, and consequently the water flows through the annular opening between *c* and *h* and, after lifting the overflow valve *k*, out of the overflow *l*. The speed of the jet gradually increases, and as soon as its momentum is sufficient, the jet forces the boiler check-valve and enters the boiler. The overflow valve *k* then closes automatically and the injector is working.

21. If the jet should break from any cause, the water will lift the overflow valve and come out of the overflow, but as soon as the momentum is sufficient again, the water will enter the boiler once more and the overflow valve will close automatically.

22. To stop the injector, the handle *a* is turned so as to screw the valve stem inwards. The steam nozzle *d*, which is movable longitudinally, remaining at rest, the valve at the end of *b* first closes the central opening in the nozzle; then, as the motion of the handle continues, the nozzle and valve stem move forwards together until the conical seat on the nozzle sits on the steam-jet guide *m*, when steam is completely shut off from the steam nozzle. To start the injector, the valve stem is slowly turned by means of the handle *a*, which first opens the central opening of the nozzle, and then, as the nozzle moves backwards with the valve stem, the other openings of the nozzle also admit steam and the injector starts.

23. Double-Tube Injectors.—Fig. 4 shows the **Hancock inspirator,** which is one of the earliest types of a double-tube injector. The term "inspirator" applied to it is merely a trade name. Steam from the boiler enters through the pipe *a* and flows through the steam nozzle *n* into the combining nozzle *o*, thereby causing water to flow up the pipe *b* into the lifting side of the instrument. The water then passes in the direction of the arrows to the forcing side of the instrument, entering at the top of the forcing tube *s*, where it is met by a jet of steam flowing

through the forcing steam nozzle *r* and is further heated and given an increased velocity. It then passes through the pipe *c* to the boiler.

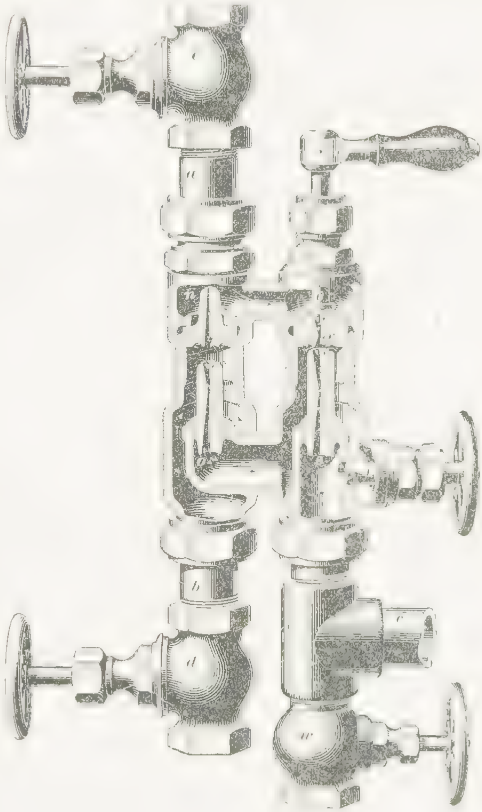
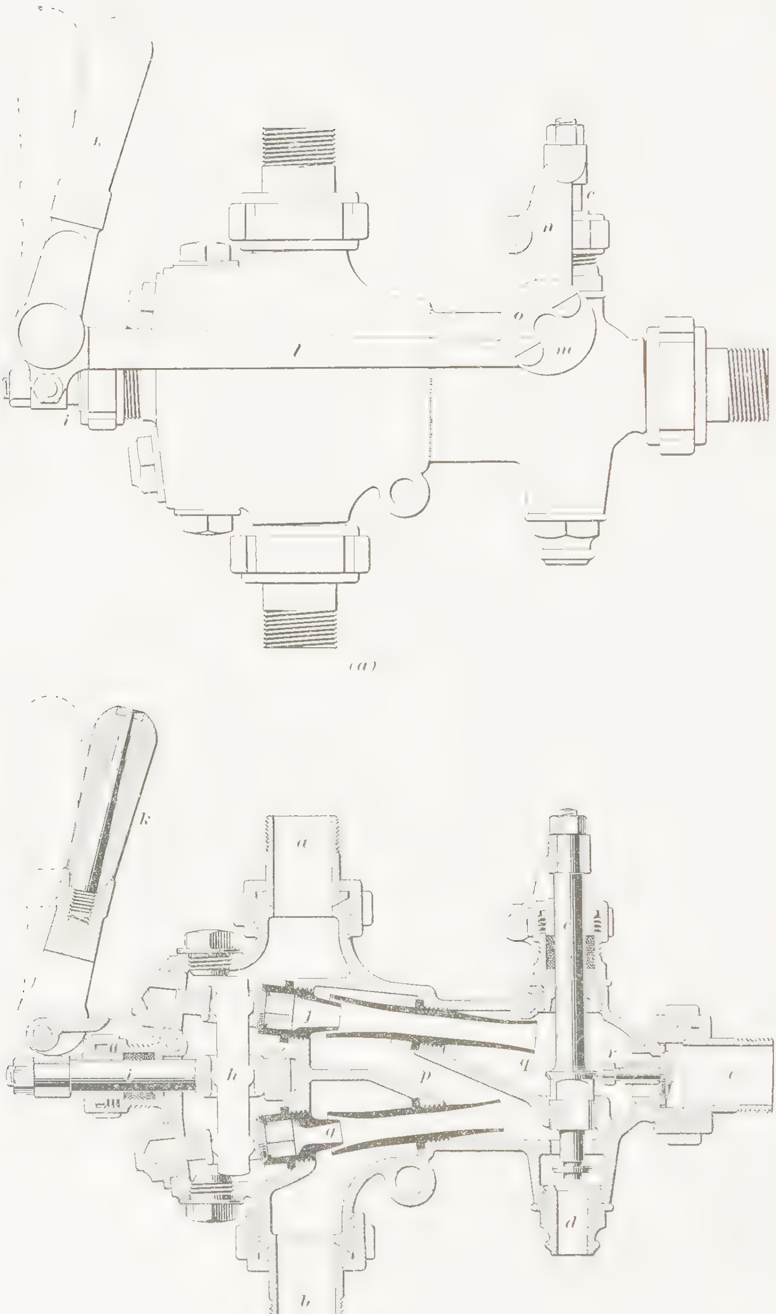


FIG. 4.

24. In order to start the instrument, the valve *v* must be closed and the overflow valves *i* and *w* opened. Next, the water valve *d* and then the steam valve *e* are opened, when the steam will rush through *n*, *o*, *i*, and *w* and out of the overflow until it creates a sufficient vacuum on the left, or lifting, side to cause the water to flow up, which will then discharge out of the overflow. As soon as the water appears, the valve *i* must be closed and the valve *v* opened. Immediately thereafter the overflow valve *w* is to be closed, when the inspirator will be working. To stop the injector, the valves *e* and *v* must be closed and *i* and *w* opened.

25. The **Korting universal double-tube injector** is shown in elevation in Fig. 5 (*a*) and in section in Fig. 5 (*b*). This injector, in common with the majority of double-tube injectors, contains a mechanically operated overflow valve, which is closed by the act of starting the injector to feed the boiler. In this injector the lower nozzles constitute the



(a) (b)

FIG. 5

lifting apparatus that delivers the water to the upper nozzles, where it is given sufficient velocity to enter the boiler. Steam enters at *a* and the water enters at *b*; the overflow *d* is closed by the valves on the stem *c*, and the water passes to the boiler through *e*. The steam nozzles *f* and *g* are closed by valves connected by means of the yoke bar *h* to the starting shaft *i*.

26. The operation of the injector is as follows: The starting handle *k* being in the position shown, the overflow valves are wide open, but the nozzles *f* and *g* are closed by their respective valves. The steam- and water-admission valves are now opened and the handle *k* is pulled over gently towards the position shown in dotted lines. This causes the starting shaft *i* and the yoke bar *h* to move in the same direction as the handle, and consequently the valves closing the nozzles *f* and *g* are opened. At the same time the overflow valves are closed slightly, the starting shaft being connected to the overflow-valve stem by links *l*, bell-cranks *m*, and links *n*. The bell-cranks have their fulcrum at *o*. The steam rushing through the lower nozzle *g* creates a partial vacuum in the water-supply pipe and causes the water to flow up, which is then delivered into the chamber *p* and passes out of the overflow. Some of the water in *p* will pass to the nozzle *f* and will deliver into the chamber *q* and thence into the overflow. In a very short time the water will be flowing freely from the overflow, and the handle is then pulled over as far as it will go, this operation opening the steam-nozzle valves to their full extent and at the same time closing the overflow openings of the chambers *p* and *q*. The injector is now working, the check-valve *r* being forced open.

27. Since the overflow outlet is positively closed, the effects of too much steam or water cannot manifest themselves by either fluid coming from the overflow. The effect of too much water will be the stopping of the injector, which can be told by the absence of vibration in the feed-pipe and the comparatively low temperature of the injector. Too

much steam manifests itself by the heating of the injector and failure to work; the remedy is either to reduce the steam supply or to increase the water supply, as the heating shows that the steam is not being condensed. These statements apply to all injectors having positively closed overflows, i. e., overflow valves so constructed that they cannot lift automatically when the injector fails to force the water into the boiler.

28. The **Monitor lifting injector** shown in Fig. 6 occupies an intermediate position between the single-tube and double-tube injectors, for while it has two sets of tubes,

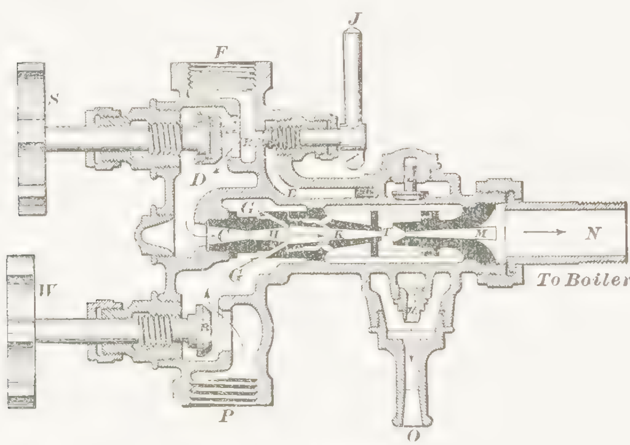


Fig. 6

the one set is used in starting the injector, but is thrown out of action as soon as the injector is working. Steam enters the injector at *F*; the water enters at *P* and passes to the boiler through the nipple *N*; the overflow is at *O*.

29. The operation is as follows: The water-admission valve *B* is first opened by turning the hand wheel *W*; the primer valve *R* is then opened by the handle *J*, thus permitting steam to flow through the passage *E* and a connection, not shown in the figure, to the nozzle *u*. From *u* the steam rushes into the overflow nozzle *O*, this nozzle in conjunction with the nozzle *u* forming the lifting part of a

double-tube injector. A passage connects the chamber surrounding *u* with the space above the overflow valve *L*. The jet of steam rushing from *u* through *O* carries with it some of the air in the chamber to which *O* is connected, thus forming a partial vacuum in the space above the overflow valve, which opens and thus allows the air in *D*, *C*, *G*, *H*, *K*, *T*, and *P* to be exhausted. The pressure of the atmosphere now forces the water up into the injector, and it finally appears at the overflow. As soon as this happens, the valve *R* is closed, which throws the priming part of the injector out of action. The steam valve *A* is now opened by turning the wheel *S*, which admits steam to the nozzles of the injector proper. At first the water will come out of the overflow, but as soon as the velocity has become high enough, it will enter the boiler, the overflow valve *L* closing automatically.

30. Exhaust-Steam Injectors.—The injectors described in Arts. **17** to **29** are intended to use live steam; there are injectors in the market, however, that make use of exhaust steam. Their principle of action is the same as that of live-steam injectors, from which they differ only in the relative proportion of the nozzles. Injectors designed to be used with exhaust steam generally will not work against boiler pressures exceeding 75 pounds; there are so-called **high-pressure exhaust-steam injectors** in the market, however, in which live steam can be introduced in order to adapt the exhaust injector to high boiler pressures.

31. Since exhaust steam is available only when the engine is running, it is necessary to furnish an exhaust injector with a live-steam connection in order that it may be used when the engine is not working. The live steam is throttled so that it will enter the injector at about the pressure of the exhaust steam. The mere addition of a live-steam connection will *not* convert an exhaust injector into a high-pressure exhaust injector, nor will an exhaust injector work when supplied with steam at full boiler pressure. Likewise, an injector designed to work with full boiler pressure will not work with exhaust steam.

DETERMINING SIZE OF INJECTORS.

32. Most engineers prefer to select a size of injector having a capacity per hour about one-half greater than the maximum evaporation per hour in order to have some reserve capacity. The maximum evaporation, when not known, may be estimated in gallons by one of the following rules, which hold good for ordinary combustion rates under natural draft:

Rule 1.—*For plain cylindrical boilers, multiply the product of the length and diameter in feet by 1.3.*

Rule 2.—*For tubular boilers, either horizontal or vertical, multiply the product of the square of the diameter in feet and the length in feet by 1.9.*

Rule 3.—*For water-tube boilers, multiply the heating surface in square feet by .4.*

Rule 4.—*For boilers not covered by the foregoing rules, multiply the grate surface in square feet by 12.*

Rule 5.—*If the coal consumption in pounds per hour is known, it may be taken as representing the number of gallons evaporated per hour.*

As there is no standard method of designating the size of an injector that is followed by all makers, such an instrument must be selected from the lists of capacities published by the different makers.

INJECTOR INSTALLATION.

33. An injector must always be placed in the position recommended by the maker, for the reason that some injectors will work well only in one position. There must always be a stop-valve in the steam-supply pipe to the injector, which should, for convenience, be placed as close to the injector as is possible. While lifting injectors, when working as such, scarcely need a stop-valve in the suction pipe, it is advisable to supply it. When the water flows to the injector under pressure, a stop-valve in the water-supply pipe is

a necessity. A stop-valve and check-valve must be placed in the feed-delivery pipe, with the stop-valve next to the boiler. The check-valve should never be omitted, even though the injector itself is supplied with one. No valve should ever be placed in the overflow pipe, nor should the overflow be connected directly to the overflow pipe, but a funnel should be placed on the latter so that the water can be seen. This direction does not apply to the inspirator or to any other injector that has a hand-operated, separate overflow valve. In the inspirator the overflow pipe is connected directly to the overflow, but the end of the pipe must be open to the air. In general, where the injector lifts water it is not advisable to have a foot-valve in the suction pipe, as it is desirable that the injector and pipe may drain itself when not in use. It is a good idea to place a strainer on the end of the suction pipe.

34. The steam for the injector must be taken from the highest part of the boiler, as it is essential to the successful working of the injector that it be supplied with dry steam. Under no consideration should the steam be taken from another steam pipe; the injector should always have its own independent steam-supply pipe. The suction pipe should be as straight as possible and must be absolutely air-tight. A very important consideration in connecting up an injector is to have the pipes cleaned by blowing them out with steam before making the connection, since quite a small bit of dirt getting into the injector will interfere seriously with its working. It is recommended to always so locate the injector that the steam pipe, suction pipe, and feed-delivery pipe will be as straight and as short as possible.

35. In some cases, especially with horizontal boilers without a dome, it is advisable to use a so-called **supplementary dome,** which is simply a vertical piece of, say, 2-inch pipe about 12 to 18 inches long; the injector steam pipe is then connected to the top of this supplementary dome.

INJECTOR TROUBLES.

36. In discussing the difficulties experienced with injectors, the suction pipe, strainer, feed-delivery pipe, and check-valve are considered as parts of the injector, since a disorder in any of these affects the work of the injector itself. In searching for the cause of a trouble, therefore, the suction and delivery pipes should be carefully inspected as well as the injector.

37. Failure to Raise Water.—The causes that prevent an injector raising water are:

1. *Suction Pipe Stopped Up.*—This is due, generally, to a clogged strainer or to the pipe itself being stopped up at some point. This prevents water from coming through and is probably the most frequent cause of an injector not priming.

In case the suction pipe is clogged, blow steam back through the pipe to force out the obstruction.

2. *Leaks in Suction Pipe.*—When this is the case, air enters and prevents the injector forming the vacuum required to raise the water. To test the suction pipe for air leaks, plug up the end and turn the full steam pressure on the pipe. The presence of leaks will then be revealed by the steam issuing therefrom. To get steam into the suction pipe, the overflow valve must be held to its seat in some convenient way, or the overflow must be closed tight in some manner. It is advisable to have the suction pipe full of water before steam is turned on, since the presence of small leaks will be revealed better by water than by steam. After testing, do not forget to clear the overflow and the end of the suction pipe.

3. *Water in the Suction Pipe Too Hot.*—In case the feed-water supply is taken from a barrel or tank and the supply is cool under normal conditions, a leaky steam valve or leaky boiler check-valve and leaky injector check-valve may be the cause of hot water or steam entering the source of supply and heating the water so hot that the injector refuses to handle it.

The reason why hot water in the suction pipe affects the operation of the injector is as follows: The pressure at which water boils depends on the pressure to which it is subjected. It has been determined by experiment that water will boil at about 380° F. under a gauge pressure of 180 pounds; at 212° F. when subjected to an atmospheric pressure of about 14.7 pounds per square inch; and at about 160° F. when in 10 inches of vacuum. This shows us that decreasing the pressure on the water lowers its boiling point. Now, when the lifting jet of an injector is turned on, a vacuum is formed in the suction pipe; and if the water there is at a temperature of 160° to 175° F., it gives off steam vapor, which fills the suction pipe and destroys the vacuum.

In case the water is too hot, cool it in any convenient manner and at the first opportunity trace out the cause and remove it. When the water in the suction pipe is very hot, but the water in the source of supply is cool, hold the overflow valve to its seat in any convenient manner or plug the overflow and open the steam valve. The steam pressure will then force the hot water out of the suction pipe. Open the overflow valve or overflow as soon as this has been done; the cool water entering the suction pipe should now be raised easily. Hot water in the suction pipe only is most likely due to a choked overflow.

4. *Obstruction in Tubes.*—There may be an obstruction in the lifting or combining tubes, or the spills (or openings) in the tubes through which the steam and water escape to the overflow may be clogged up with dirt or lime. In either case, the free passage of the steam to the overflow will be interfered with, and, consequently, a steam pressure instead of a vacuum will be formed in the suction pipe, the extent of the pressure depending on the amount of obstruction.

38. Injector Primes, But Will Not Force.—In some cases an injector will lift water, but will not force it into the boiler; or it may force part of it into the boiler and the rest out of the overflow. When it fails to force, the trouble may be due to one or the other of the following causes:

1. *Choked Suction Pipe or Strainer.*—If the suction pipe or the strainer is partially choked, the injector, in either case, will be prevented from lifting sufficient water to condense all the steam issuing from the steam valve. The uncondensed steam, therefore, will gradually decrease the vacuum in the combining tube until it is reduced so much that the injector will break. It is to be remembered that when the injector is operating, it is the vacuum *in the combining tube* that causes the water to be raised.

The remedy in case the supply valve is partially closed is obvious. In the case of choked suction pipe, blow out the obstruction.

2. *Suction Pipe Leaking.*—The leak may not be sufficient to entirely prevent the injector lifting water, but the quantity lifted may be insufficient to condense all the steam, which, therefore, destroys the vacuum in the combining tube. A slight leak may exist that will simply cut down the capacity of the injector. In such a case an automatic injector will work noisily, on account of the overflow valve seating and unseating itself as the pressure in the combining tube varies, due to the leak.

3. *Boiler Check-Valve Stuck Shut.*—If completely closed, the injector may or may not continue to raise water and force it out of the overflow—it depends on the design of the injector.

If the boiler check is partly open, the injector will force some of the water into the boiler and the remainder out of the overflow. In case the check-valve cannot be opened wide, water may be saved by throttling both steam and water until the overflow diminishes, or, if possible, ceases. The steam should be throttled at the valve in the boiler steam connection, however, and not at the steam valve of the injector, as throttling tends to superheat the steam, and an injector will not work as satisfactorily with superheated steam as with saturated steam. By throttling the steam at the boiler, the excess of heat due to this throttling will be lost before the steam reaches the injector.

If a check-valve sticks, it can sometimes be made to work again by tapping *lightly* on the cap or on the bottom of the valve case.

4. *Obstruction in Delivery Tube.*—Any obstruction in the delivery tube, such as cotton waste, scale, or coal, will cause a heavy waste of water from the overflow. To remedy this, the tube will probably have to be removed and cleaned.

5. *Leaky Overflow Valve.*—This not only diminishes the capacity of the injector, but it allows air to be drawn into the boiler; and if the leak is sufficiently great, it will destroy the vacuum in the combining tube and prevent the injector operating. A leaky overflow valve is indicated by the boiler check chattering on its seat. To remedy this defect, grind the valve on its seat until it forms a tight joint.

6. *Injector Choked With Lime.*—It is essential to the proper working of an injector that the interior of the tubes should be perfectly smooth and of the proper bore. As in course of time they lime up, the capacity of the injector decreases until, finally, it refuses to work at all. If the water used is very bad, it becomes necessary to frequently cleanse the tubes of the accumulated lime. This may be accomplished by putting the parts in an acid bath, allowing the acid to remove the scale. The bath should consist of 1 part of muriatic acid to 10 parts of water. The tubes should be removed from it as soon as the gas bubbles cease to be given off. The acid combines with the lime and forms a gas, and as long as there is lime to combine with, it will not attack the copper in the tubes. After the lime has all combined, however, the acid will attack the tubes, with the result that the inner surface will become pitted and rough, which will affect the working of the injector.

ADVANTAGES AND DISADVANTAGES.

39. The advantages of the injector as a boiler feeding apparatus are its cheapness, as compared with a pump of equal capacity; it occupies but little space; the repair bills

are low, owing to the absence of moving parts; no exhaust piping is required, as with a steam pump; it delivers hot water to the boiler. The disadvantages of the injector are that it will not start with a steam pressure less than that for which it is designed, and that it will stand but little abuse, being poorly adapted for handling water containing grit or other matter liable to cut the nozzles.

ECONOMY OF INJECTOR.

40. Such economy as is derived from the use of an injector is not chargeable to its economy in the use of steam, for it uses as much steam as a fairly good steam pump, but rather to the fact of the fundamental principle of its operation insuring the delivery of hot feedwater to the boiler. The introduction of hot feedwater has a marked effect upon the repair bills and tends to increase the life of a boiler by reason of the diminishing of the stresses incidental to expansion and contraction.

GRAVITY-FEED APPARATUS.

41. When the water supply has a pressure sufficient to elevate it a few feet above the boiler, or when the source of supply is located above the boiler, a **gravity feed** may be made use of, dispensing with pumps or injectors.

42. The simplest form of a gravity-feed apparatus consists of a closed vessel placed about 6 feet above the boiler and connected at the top by a steam pipe to the boiler, while at the bottom it connects to the feedpipe. A pipe also connects the closed vessel to the source of supply. In operation, the vessel is first nearly filled with water, and steam from the boiler is then admitted on top of the water, thus submitting it to the full boiler pressure. Then, by reason of the vessel being elevated above the boiler, there is sufficient hydrostatic head for the water to flow into the boiler. Since the water is in direct contact with live steam, it will

be heated to a fairly high temperature. Whenever the vessel is empty, the steam is shut off and water run in again, when the operation can be repeated.

43. A more elaborate gravity-feed system uses a steam trap and has the advantage of an automatic refilling of the trap and, consequently, fairly constant feed, which is adjustable to suit the demand. This system is almost invariably used to return the water of condensation from steam-heating systems to the boiler, and when not thus employed may be used for feeding water coming from some other source.

44. Fig. 7 illustrates the **Bundy steam trap,** which is largely used for gravity-feed systems. The trap consists of a body *a* that receives the water and that is pivoted at *b* and *c* to the stationary part of the trap. The water enters the trap at *b* and fills the part *a* until the weight of the contained water is sufficient to overbalance the weight *d*, when *a* sinks downwards until it comes against the guide *g*. This downward motion causes the lug *e* to engage the upper nuts of the steam valve *f*, opening the latter and at the same time closing a small air valve placed below *f*, thus admitting steam at full boiler pressure on top of the water, which now flows to the boiler by gravity, being able to enter the latter by reason of its hydrostatic head. As soon as the trap is emptied enough, the weight *d* pulls the trap body to its upper position again, which closes the steam valve and allows air to enter the trap body. The operation is now repeated.

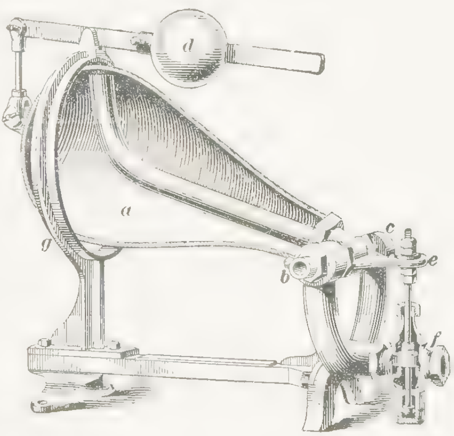


FIG. 7.

45. The piping up of a gravity-feed system using the Bundy trap is shown in Fig. 8. In case of a heating or steam-pipe drainage system, the different drains lead to a receiver *a*, whence the water passes to the trap *d* through the pipe *b* and check-valve *c*. The feedwater leaves the

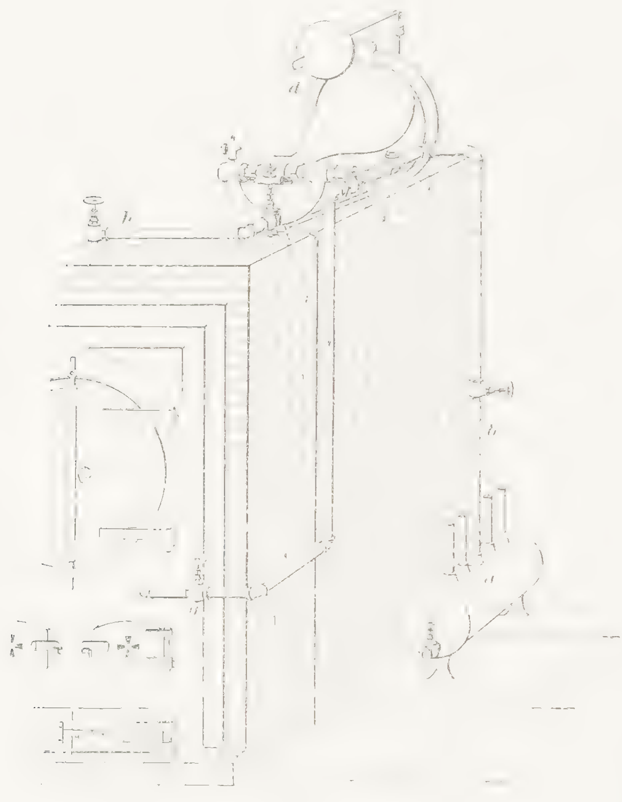


FIG. 8.

trap and passes to the boiler through the feedpipe *e*, the check-valve *f*, and the globe valve *g*. The check-valve *c* prevents the water or live steam in the trap blowing back into the receiver, while the check-valve *f* prevents the water in the boiler backing up into the trap. Steam is admitted to the trap through the pipe *h*; a vent pipe *i* is usually attached

to the air valve and conveys the steam left in the trap after it is emptied of water to the ash-pit.

46. When no water of condensation from a heating system is available, or when that supply is insufficient, a connection may be made to a water-service pipe, arranging the pipe connections and valves in such a manner that water may be taken at will from either source of supply, in case there is more than one. When there is not sufficient water pressure available to make the water enter the trap on top of the boiler, a trap may be placed at the point where water will flow into it. This trap may then be made to discharge into another one placed on top of the boiler, using steam from the boiler as a motive force.

AUTOMATIC FEEDWATER REGULATION.

47. It has long been recognized that it is desirable to keep the water level in the boiler steady, this being conducive to a steady steam pressure and incidental economy in the use of fuel. There are a number of devices in the market that control the steam-supply valve of a pump or injector or the feedpipe valve in such a manner that water will be supplied to the boiler at the same rate at which it is evaporated, thus keeping the water level steady. Incidentally, these devices tend to prevent the water becoming low enough to be dangerous or high enough to be carried over to the engine.

48. Fig. 9 shows the **Vigilant feedwater regulator.** It consists essentially of a water column *a* carrying on top an oscillating hollow lever *b* pivoted at its center and carrying a hollow sphere *c* at one end and a counterweight *d* at the other end. The sphere *c* is in communication with the inside of the water column through the hollow lever *b*, its trunnions, a passage in the yoke *e*, and a pipe *f* extending inside the column down to the normal water level. A so-called condenser *g* is in communication with the water column and has a valve *h* in the pipe *i* leading to the

regulator valve *k*, which, in case of a pump, is placed in the feed-delivery pipe, and in case of an injector, in the steam pipe. A relief valve *l* relieves the pressure in the pipe *i* whenever it is opened. The valves *h* and *l* are opened through collars on the lever *b* coming into contact with the

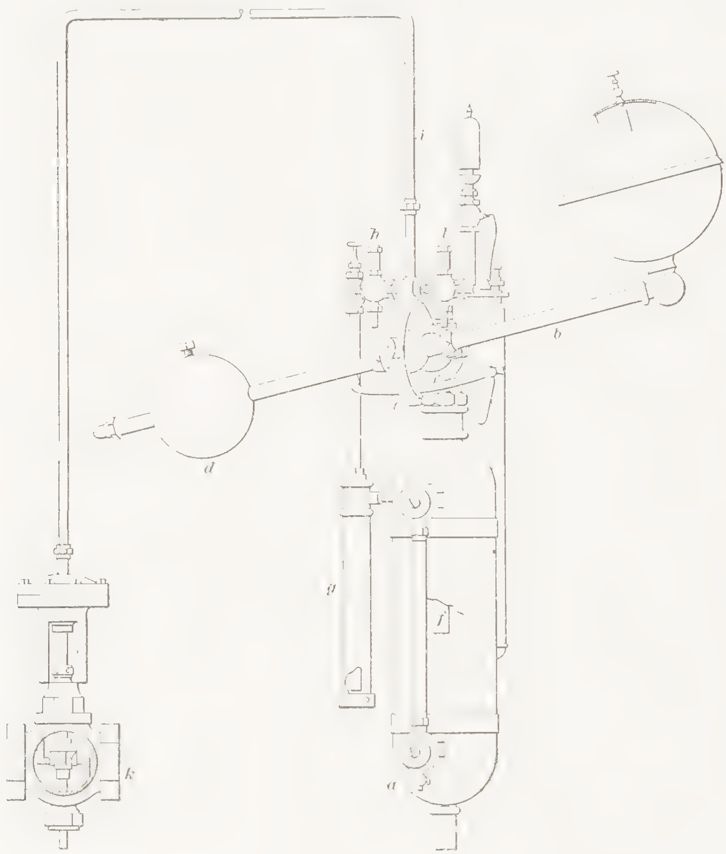


Fig. 9.

valve stems. The regulator valve *k* has a large diaphragm in its upper part, which is connected to the stem of the valve, and when the valve *h* is open, is subjected to practically the full boiler pressure, which tends to close the regulator valve.

49. The operation is as follows: when the water in the boiler has sunk enough to uncover the lower opening of the pipe *f*, the sphere *c* is filled with steam, and the balance weight *d* holds the lever *b* in the position shown, in which the relief valve *l* is forced open. There being no pressure on the back of the regulator valve, the pressure in the feedpipe holds it open and the water enters the boiler until the water in the column seals the mouth of the pipe *f*. Water is now forced through *f*, *c*, and *b* into *c* until the weight of the water in *c* is sufficient to overbalance the weight *d*, when the lever oscillates to a horizontal position, and in so doing closes the relief valve *l* and opens the pressure valve *h*. The diaphragm in *k* is now subjected to the full boiler pressure and the valve in the regulator closes, thus shutting off the feed, which remains closed until the water level in the column has again sunk below the mouth of *f*. Steam then enters *c* and allows the water to drain out of it; in consequence, the weight *d* soon pulls the lever *b* to its inclined position, which opens the relief valve *l* and closes the pressure valve *h*, thus relieving the pressure on top of the diaphragm. The pressure in the feedpipe now opens the regulator valve and water again enters the boiler.

FEEDWATER PROBLEMS.

INCRUSTATION IN GENERAL.

IMPURITIES FOUND IN FEEDWATER.

50. Incrustation is a deposit that is formed on the plates and tubes of a boiler; it is caused by impurities in the water that are left behind in the boiler.

If the water used in a boiler were perfectly pure there would be, of course, no trouble from incrustation. Unfortunately, however, in passing through the soil, water dissolves certain mineral substances, the most important of

which are carbonate of lime, which is the same thing as limestone or marble, and sulphate of lime, which is the same as plaster of Paris. Other substances frequently present in small quantities are chloride of sodium (common salt) and chloride of magnesium. It also often contains other troublesome substances.

51. Carbonate of lime will not dissolve in pure water, but will dissolve in water that contains carbonic-acid gas. It becomes insoluble and is precipitated in the solid form when the water is heated to about 212°, the carbonic-acid gas being driven off by the heat.

52. Sulphate of lime dissolves readily in cold water, but not in hot water. It precipitates in the solid form when the water is heated to about 290°, corresponding to a gauge pressure of 45 pounds.

53. Chloride of sodium will not be precipitated by the action of heat unless the water has become saturated with it. Since it generally is present in but very small quantities in fresh water, it will take a very long time for the water in a boiler to become troublesome, and with the ordinary blowing down of a boiler once a week or every 2 weeks, there is little danger of the water becoming saturated with it. Consequently, it is one of the least troublesome scale-forming substances contained in fresh water.

54. Chloride of magnesium is one of the worst impurities in water intended for boilers, for while not dangerous as long as the water is cold, it makes the water corrosive when heated, and when present in large quantities, it becomes dangerously corrosive, attacking the metal of the boiler and rapidly corroding it.

55. Carbonate of magnesia and **sulphate of magnesia** are not very troublesome constituents of feedwater.

56. Organic matter by itself may or may not cause the water to become corrosive, but will often cause foaming; when it is present in small quantities in water containing

carbonate or sulphate of lime, or both, it usually serves to keep the deposits from becoming hard.

57. Earthy matter, like organic matter, is not dissolved in the water, but is in mechanical suspension. It is very objectionable, especially when the earthy matter is clay, and when other scale-forming substances are present, is liable to form a hard scale resembling Portland cement.

58. Acids, such as sulphuric acid, nitric acid, tannic acid, and acetic acid, are often present in the feedwater. The sulphuric acid is the most dangerous one of these acids, attacking the metal of which the boiler is composed and corroding it very rapidly. The other acids, while not so violent in their action as the sulphuric acid, are also dangerous, and water containing either one should be neutralized when it must be used.

FORMATION OF SCALE.

59. The small solid particles, due to precipitation of substances in solution or matter in mechanical suspension, remain for a time suspended in the water, especially the carbonate of lime that for some time after precipitation floats on the surface of the water. These particles will gradually settle on the plates, tubes, and other internal surfaces. A large part of the impurities will be carried by the circulation of the water to the most quiet part of the boiler and there settle and form a scale. In a few weeks, if no means of prevention are used, the inner parts of the boiler will be covered with a crust from $\frac{1}{16}$ to $\frac{1}{2}$ inch in thickness.

60. A scale $\frac{1}{32}$ inch or less thick is thought by many to be an advantage, since it protects the plates from the corrosive actions of acids in the water. When, however, the scale becomes $\frac{1}{4}$ inch thick or more, heat is transmitted through the plates and tubes with difficulty, more fuel is required, and there is danger of overheating the plates. No reliable tests have ever been made as to the loss in efficiency due to different thicknesses of scale, and while it is generally admitted that there is some loss, its magnitude is believed

by many of the best engineers to be comparatively small. The chief danger from a heavy incrustation is the danger of overheating the plates and tubes; it also prevents a proper examination of the inside of the boiler, since it may hide a dangerously corroded piece of plate or a defective rivet head that would otherwise be discovered. Incrustation in many cases has led to danger and disaster by stopping up the feedpipe, the blow-off pipe, or the connections to the gauge glass.

The carbonate of lime forms a soft, muddy scale, which when dry becomes fluffy and flour-like. This scale may be easily swept or washed out of the boiler by a hose, provided it is not baked hard and fast. A carbonate scale is much harder to deal with when grease is allowed to enter the boiler. The grease settles and mixes with the floury scale, making a spongy crust that remains in contact with the plates, being too heavy to be carried off by the natural circulation of the water. Very many cases of overheated and burned plates are the direct result of allowing grease or animal oil to enter the boiler.

The sulphate of lime forms a scale that soon bakes to the plates. In addition to the scales above mentioned, a large amount of mud and earthy matter may be deposited by the use of dirty or muddy water.

REMOVAL OF INCRUSTATION.

61. When incrustation has once been formed, it cannot very readily be dissolved again by any chemicals, although some substances seem to soften and aid in detaching it. Of these, kerosene oil has met with much favor. Its action appears to be mechanical rather than chemical, the oil penetrating or soaking through the scale and softening and loosening it. It is somewhat useful, too, in preventing the formation of a scale, enveloping the fine particles of the scale-forming substances that, after precipitation, first float on the surface of the water for a little while. It seems that this prevents the particles from adhering firmly to one

another and to the metal when they finally settle. Engineers are divided in their opinion as to whether kerosene will not do more harm than good, many granting its efficiency as a preventative agent and its assistance in softening and loosening scale, but claiming that it leaves the metal entirely unprotected and thus permits of pitting and corrosion.

62. A hard scale once formed is generally removed by chipping it off with scaling hammers and scaling bars; soft scale can be largely removed during running by a periodic use of the bottom and surface blow-off, and the remainder can usually be washed out and raked out when the boiler is blown out and opened. In order to prevent the scale-forming substances deposited on the metal baking hard, it is advisable to let the boiler cool down slowly until entirely cold preparatory to blowing off, whenever circumstances permit this to be done. This cooling down process will generally take from 24 to 36 hours.

63. Mud and earthy matter by itself will not form any hard scale, but will often do so when carbonate of lime and sulphate of lime are present. An accumulation of such matter can be prevented, and most of the matter present can be removed, by a periodic use of the bottom blow-off, removing the remainder whenever the boiler is opened.

PREVENTION OF INCRUSTATION AND CORROSION.

64. Incrustation can best be prevented by purifying the feedwater prior to entering the boiler, and can be fairly satisfactorily prevented by a chemical treatment of the water in case purification prior to the water entering the boiler is not resorted to. When the water contains large quantities of substances that float on the surface, mechanical means may be resorted to, using either the surface blow-off at frequent intervals or some equivalent skimming device. Corrosion is prevented by neutralizing the corrosive acids by an alkali; corrosion due to a perfectly fresh

TABLE II.

SCALE-FORMING SUBSTANCES AND THEIR REMEDIES.

Troublesome Substance.	Trouble.	Remedy or Palliation.
Sediment, mud, clay, etc.	Incrustation.	Filtration. Blowing-off.
Readily soluble salts.	Incrustation.	Blowing-off.
Bicarbonates of lime, magnesia, iron.	Incrustation.	Heating feed. Addition of caustic soda, lime, or magnesia.
Sulphate of lime.	Incrustation.	Addition of carbonate of soda or barium chloride.
Chloride and sulphate of magnesium.	Corrosion.	Addition of carbonate of soda, etc.
Carbonate of soda in large amounts.	Priming.	Addition of barium chloride.
Acid (in mine water).	Corrosion.	Alkali.
Dissolved carbonic acid and oxygen.	Corrosion.	Heating feed. Addition of caustic soda, slaked lime, etc.
Grease (from condensed water).	Corrosion.	Slaked lime and filtering. Carbonate of soda. Substitute mineral oil.
Organic matter (sewage).	Priming.	Precipitate with alum or chloride of iron and filter.
Organic matter.	Corrosion.	Same as last.

water can be prevented by giving a protective coating to the metal, which may be a thick red-lead paint made up with boiled linseed oil, or a thin coating of scale.

65. Sometimes organic substances containing tannic acid, such as oak bark, hemlock, or sumac, are used to loosen or prevent scale. They are liable to injure the plates by corrosion and hence should not be used.

66. Table II gives a list of troublesome scale-forming substances and their remedies.

67. **Zinc** is largely used in marine boilers for the prevention of both incrustation and corrosion. The scale may acquire thickness and hardness, but can easily be removed from the plates. It is supposed that the zinc in connection with the iron of the plates keeps up a feeble galvanic action, and that the hydrogen liberated at the surface of the plate by this action prevents the incrustation from adhering to it. The zinc is distributed through the boiler in the form of slabs. About 1 square inch of zinc surface should be supplied for every 50 pounds of water.

PURIFICATION OF FEEDWATER.

MEANS OF PURIFICATION.

68. Water intended for boilers may be purified by settlement, by filtration, by chemical means, and by heat. Filtration will remove impurities in mechanical suspension, such as oil and grease, and earthy matter, but will not remove substances dissolved in the water. Chemical treatment of the water will render the scale-forming substances and corrosive acids harmless, and may be applied either before or after the water enters the boilers, preferably the former. Purification by heat is based on the fact that most of the scale-forming substances become insoluble and precipitate when the water containing them in solution is heated to a high temperature.

PURIFICATION BY SETTLEMENT.

69. For feedwater containing much matter in mechanical suspension, one of the simplest methods of purifying it is to provide a relatively large reservoir or a large tank for small steam plants, where the impurities can settle to the bottom. While this method is fairly satisfactory, as far as earthy matter is concerned, it will not clear the water of finely divided organic matter, which is usually lighter than the water and often so finely divided as to be almost dissolved in it.

PURIFICATION BY FILTRATION.

70. Organic and earthy matter in mechanical suspension is most satisfactorily removed by a filter, passing the water through layers of sand, gravel, hay, or equivalent substances, or through layers of cloth. Hay and cloth are of service especially where the feedwater contains oil or grease, as is the case where a surface condenser is used and the condensed steam is used over again.

71. A hay filter is extremely simple, consisting of a wooden tank open on top and divided into two compartments by a partition perforated near the bottom. The one compartment is filled with hay, the feedwater from the hot well being discharged into the top of it. The hay quite effectually removes the oil and grease from the water, which is taken from the second compartment by the feed-pump or injector. The hay will have to be renewed quite frequently.

72. There are a number of cloth filters in use in condensing plants. One of these, known as the **Ross feedwater filter,** designed to remove oil and grease from the feedwater of a surface-condensing engine, is shown in Fig. 10. The water coming from the feed-pump enters at *a* and passes into the filtering chamber *b*. It cannot leave this filtering chamber without passing through the filter *c*, which consists of light circular bronze sections of open

latticework held together by long bolts and covered by toweling. This material is technically known as "linen terry," and popularly as "Turkish toweling." The toweling is made up in the shape of a bag somewhat larger than the spider; it is drawn over it and drawn down between each of the sections by a string wound around it. The feedwater slowly passes through the filtering material into the interior of the filter; it then goes through the left-hand opening of the filtering chamber and through the valve *d* into the feedpipe

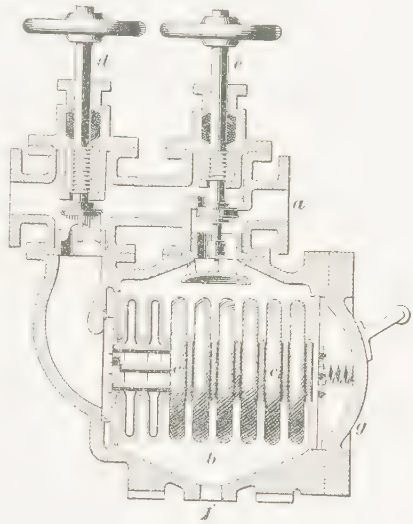


FIG. 10

again and thence to the boiler. The foreign matter filtered from the water accumulates on the filtering material, and in course of time offers considerable resistance to the passage of the water. This resistance is shown by the difference in reading of two pressure gauges. One of these is connected to the chamber *b* and the other to the left-hand passage. When this difference amounts to 3 pounds, the filter is in need of cleaning. To clean the filter, close valves *e* and *d* and open the drain at *f*. Now open valve *e* a little. A current of water will then flow around the filter and out of the drain, washing the outside of the toweling. Next, close valve *e* again and open valve *d*. Then, the drain being open, a current of water will flow through the filter in a direction opposite to that in which the water passes through it when filtering. The water flowing in a reverse direction tends to loosen the foreign matter adhering to the outside of the filter. To start the filter again, open valves *e* and *d* and close the drain. Should it be found that the washing

of the filter, as explained above, is insufficient to clean it, a new filter must be inserted. To do this, close valves *c* and *d* and open the drain. The water from the feed-pump will now pass directly to the boiler, the screwing down of the valve *c* to close the opening to the filter chamber opening a by-pass, as shown. The cover *g* can now be removed and a new filter inserted.

73. An **Edmiston feedwater filter** is shown in Fig. 11 It consists of a vessel *a*, divided by perforated plates *b*, *b*, covered with coarsely woven cloth, into two chambers. The feedwater is admitted to the chamber *a′*, and cannot reach the chamber *a″* except by passing through the filtering

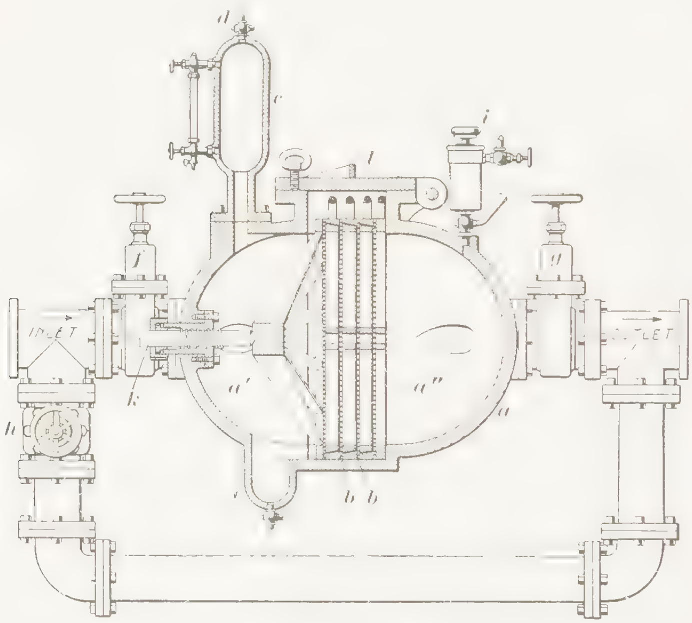


FIG. 11.

cloth. The oil and other floating impurities rise into the scum chamber *c*, whence they are removed periodically by opening the blow-off *d*. The heavier impurities settle into the pocket *e*, which is provided with a blow-off cock. A pressure gauge is attached to the chamber; when this gauge

indicates more than 5 pounds pressure in the chamber in excess of that in the boiler, it shows that the strainer is clogged and must be cleaned. This is done by opening the by-pass valve *h* and closing the valves *f* and *g*, thus cutting the filter out of the feedpipe. The soda cup *i* is now filled with soda and steam turned on, thus boiling out the filter. The soda dissolves the grease and the matter in the filter can then be blown out. If boiling out fails to clear the filter, the filtering diaphragms must be removed and new ones substituted. To do this, first of all cut the filter out of the feedpipe, letting the feedwater go through the by-pass. Then loosen the setscrew *k*. Now open the hinged door *l*. The diaphragms can then be readily removed.

PURIFICATION BY CHEMICALS.

74. Action of Different Chemicals.—Chemical purification may take place before or after the water enters the boiler, the former method being somewhat more expensive on account of the reservoirs that must be constructed. However, the purification is better carried out before the water enters the boiler for the reason that the amount of impurities entering the boiler will be greatly reduced.

75. The chemical process to be adopted depends on the substances present in the water. When the water contains only carbonate of lime, it may be treated with slaked quicklime, using 28 grains of lime for every 50 grains of carbonate of lime present in the water, the quicklime precipitating the carbonate of lime and being transformed into carbonate of lime itself during the process. When this process is carried out outside the boiler and the water cleared by settlement and filtration, very little carbonate of lime and matter in mechanical suspension will enter the boiler. This process was devised by Doctor Clark.

76. Water containing carbonate of lime may also be treated with caustic soda, which precipitates the carbonate of lime and leaves carbonate of soda, which is harmless.

For every 100 grains of carbonate of lime 80 grains of caustic soda should be added.

77. Sal ammoniac is sometimes added to water containing carbonate of lime and will cause the latter to precipitate. Its use is not advisable, however, on account of the danger of the formation of hydrochloric acid, which will attack the boiler. The formation of this acid is due to an excessive quantity of the sal ammoniac having been used.

78. While slaked lime will precipitate carbonate of lime, it will have no effect on sulphate of lime, and water containing the latter, either alone or in conjunction with carbonate of lime, must be treated with other chemicals. The most available ones for water containing both are carbonate of soda and caustic soda. These are often fed into the boiler and will precipitate the carbonate of lime and sulphate of lime there, requiring the sediment to be blown out or otherwise removed periodically.

79. The action of caustic soda on carbonate of lime and sulphate of lime in water containing both of these ingredients is as follows: The soda precipitates the carbonate of lime, and in so doing carbonate of soda is formed, which, in turn, combines with the sulphate of lime, precipitating it in the form of carbonate of lime, and in so doing forming sulphate of soda, which is very soluble and harmless and may long be neglected.

80. When treating water containing carbonate of lime and sulphate of lime, caustic soda may be used either by itself or in combination with carbonate of soda, depending on the relative proportions of carbonate of lime and sulphate of lime present in the water. The amount of caustic soda or carbonate of soda to be used per gallon of feedwater can be found as follows:

Rule 6.—*Multiply the number of grains of carbonate of lime per gallon by 1.36. If this product is greater than the number of grains of sulphate of lime per gallon, only caustic*

soda is to be used. To find the quantity of caustic soda required per gallon, multiply the number of grains of carbonate of lime in a gallon by .8.

Rule 7.—*Multiply the number of grains of carbonate of lime per gallon by 1.36. If this product is less than the number of grains of sulphate of lime per gallon, take the difference and multiply it by .78 to obtain the number of grains of carbonate of soda required per gallon. To find the amount of caustic soda required per gallon, multiply the number of grains of carbonate of lime in a gallon by .8.*

Example.—A quantitative analysis of a certain feedwater shows it to contain 23 grains of sulphate of lime and 14 grains of carbonate of lime per gallon. How much caustic soda and carbonate of soda should be used per gallon to precipitate the scale-forming substances?

Solution.—By rule **6**, $14 \times 1.36 = 19$ grains. Since this product is less than the number of grains of sulphate of lime per gallon, rule **7** is to be used. Applying rule **7**, we get $(23 - 19) \times .78 = 3.12$ gr. of carbonate of soda, and $14 \times .8 = 11.2$ gr. of caustic soda. Ans.

81. Water containing sulphate of lime, but no carbonate of lime, may be treated with carbonate of soda. The amount of the latter that is required per gallon to precipitate the sulphate of lime is found by multiplying the number of grains per gallon by .78.

82. When using soda, it is well to keep in mind that it will not remove deposited lime from the inside of a boiler. All that the soda can do is to facilitate the separating of the lime, i. e., cause it to deposit in a soft state. This sediment must be removed periodically.

83. For decomposing sulphate of lime, tribasic sodium phosphate, more commonly known as trisodium phosphate, is often used. This is claimed to act on the sulphate of lime, forming sulphate of sodium and phosphate of lime, the former of which remains soluble and is harmless, and the latter of which is a loose, easily removed deposit. Trisodium phosphate also acts on carbonate of lime and carbonate of magnesia, forming phosphate of lime and phosphate of magnesia, at the same time neutralizing the carbonic acid

released from the carbonate of lime and magnesia, and the sulphuric acid released from the sulphates.

84. Acid water can be neutralized by means of an alkali, soda probably being the best one. The amount of soda to be used can best be found by trial, adding soda until the water will turn red litmus paper blue.

85. Example of a Purification Plant.—Fig. 12 shows the general arrangement of a purification plant for a boiler

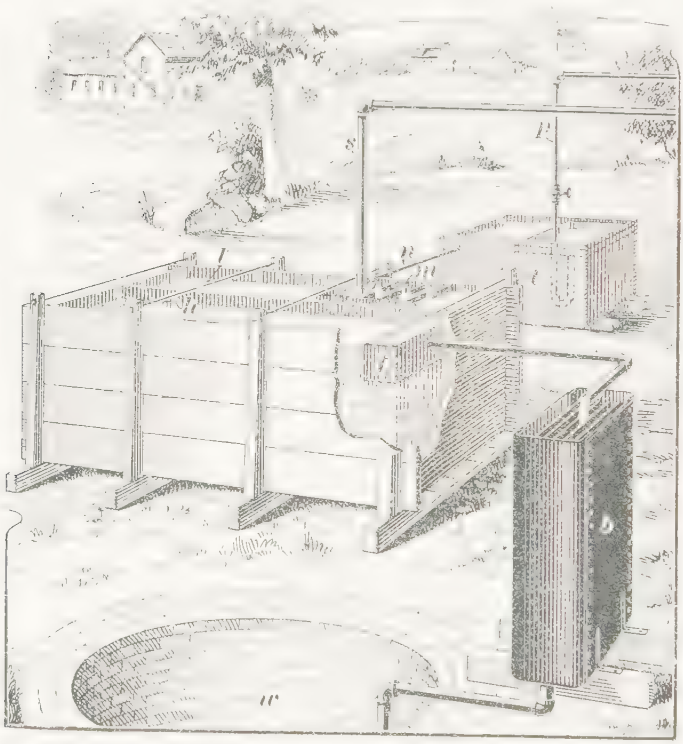


FIG. 12

plant using 60,000 pounds of water per day. The apparatus consists of a tank *t*, about 4 ft. × 3 ft. and 3 feet deep, in which lime or soda is dissolved upon a perforated tray, the

water-supply pipe *p* directing a jet upwards against the tray bottom. The tray is about 3 ft. × 2 ft. and 6 inches deep. From this tank a pipe *n* conveys the lime to a mixing tray *m*, which is the same size as the lime tray, where the lime water is mixed with the feedwater coming through the pipe *s* by passing a series of obstructing plates. From the mixer the mixed lime and water flows into the first division of the large settling tank *l*, which division is about 12 ft. × 4 ft. by 3 feet deep, the second division having the same size. The only communication between the two halves is by a small surface hole *h*, and from the second division of the large tank the final outlet is taken from the surface by the skimming trough *f*, 4 ft. × 6 ft. × 6 in. deep, and flows thence to the box *b* about 2 ft. × 1 ft. 6 in. × 4 ft. deep. This box is loosely divided into several vertical divisions by perforated plates that serve to support several flat-woven filtering bags, and when past this filter, the purified water passes off to the well *w*. By reason of the slow motion brought about by the large cross-section of the settling tank and the removal of the water at the surface by the hole *h* and trough *f*, only a small quantity of the finest material needs arresting at the filters. The bulk of the deposit will occur in the first settling tank, and when the amount of water treated exceeds the settling power, the tanks require supplementing by an additional length.

SKIMMING APPARATUS.

86. On account of the fact that some of the impurities contained in feedwater will float on the surface of the water for some time, a surface blow-off is often fitted, which serves to remove these floating impurities periodically. In order to collect the scum, a spoon-shaped receptacle or a shallow trough may be placed in the boiler, with its top 2 or 3 inches below the normal water level, and the surface blow-off pipe connected to the bottom. These devices are good so far as they go, but between times of using some of the scum settles and finds its way to the boiler bottom.

87. There is a continuous skimmer that requires very little waste of water in blowing out and the action of which is based on known physical laws. This is the **Hotchkiss boiler cleaner** shown in Fig. 13.

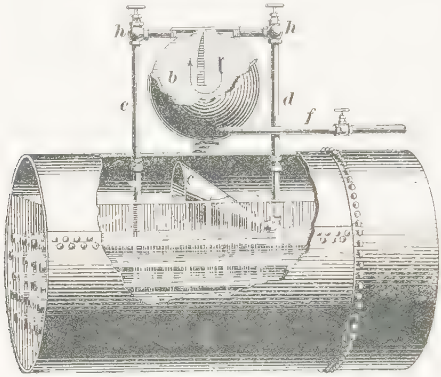


FIG. 13.

It consists of a globular vessel *b* placed above the boiler and with an internal diaphragm, as shown. Connected with it is a funnel *c*, so placed inside the boiler as to draw in water anywhere from the lowest level to the highest. Water enters this funnel and rises up the pipe *d* into the globe and there slowly circulates, depositing its contained mud and passing out to the boiler bottom by the pipe *c*, valves *h* serving to moderate the velocity of flow. The mud settles at the bottom of the globe and is blown out through the pipe *f*.

The question may be asked, how is circulation brought about? In a boiler the water that enters at *c* is not boiling but is ready to boil on the addition of another fraction of heat or the subtraction of an ounce of pressure. In the pipe *d* there is less pressure than in the boiler, by reason of the head of water above the water level in the boiler, and therefore water that enters *c* as water is turned partially into steam as it ascends, and when it enters the globe, some of it again goes back to water, because there is a slight cooling by radiation. The net result is that of the two pipes *c* and *d*, the pipe *d* contains a greater proportion of steam than does the pipe *c*, and is therefore of less density, and the downward pull in the heavier column in *c* causes circulation. The water in the boiler is, therefore, in continual circulation through *b*, entering laden with lime and leaving more or less cleared of it, and the result is simply that mud

is found in *b* instead of in the boiler. The economy that results is that due first to the ability to use the blow-off *f* only sufficiently to blow out mud without water, and secondly to the avoidance of cleaning the boiler, and finally to the reduction in repairs and saving in fuel from increased efficiency of heating surface.

88. This continuous cleaner works best with soda, which should be fed into the boiler. Where magnesium salts are present in the water, they form a scum that will not sink. In this case the globe *b* is placed upside down and all dirt then *rises* to the blow-out *f*, which is then at the upper side of the ball. In this way the oil that has got in through a surface condenser is discharged. When both sinking and floating impurities are to be dealt with, the globe is placed on its side with the diaphragm vertical and a double blow-out is used, one above for oil or magnesia froth and one below for lime, both being connected with the same blow-off.

PURIFICATION BY HEAT.

89. It was mentioned in Arts. **51** and **52** that carbonate of lime and sulphate of lime become insoluble if the water is heated, the former precipitating at about 212° F. and the latter at about 290°. This fact is taken advantage of in devices that may be called **combined feedwater heaters and purifiers**; since they generally use live steam, they are also called **live-steam feedwater heaters.** Since no feedwater heater can effect a direct saving in fuel except when the heat is taken from a source of waste, it follows that a live-steam feedwater heater can affect the fuel consumption but indirectly. This it does by largely preventing the accumulation of scale in the boiler and the attendant loss in economy due to the lowering of the rate of heat transmission through a plate heavily covered with incrustation. A live-steam feedwater heater, by reason of the delivery of very hot water, also tends to reduce repair bills and increases the life of a boiler on account of the decrease in expansion and contraction.

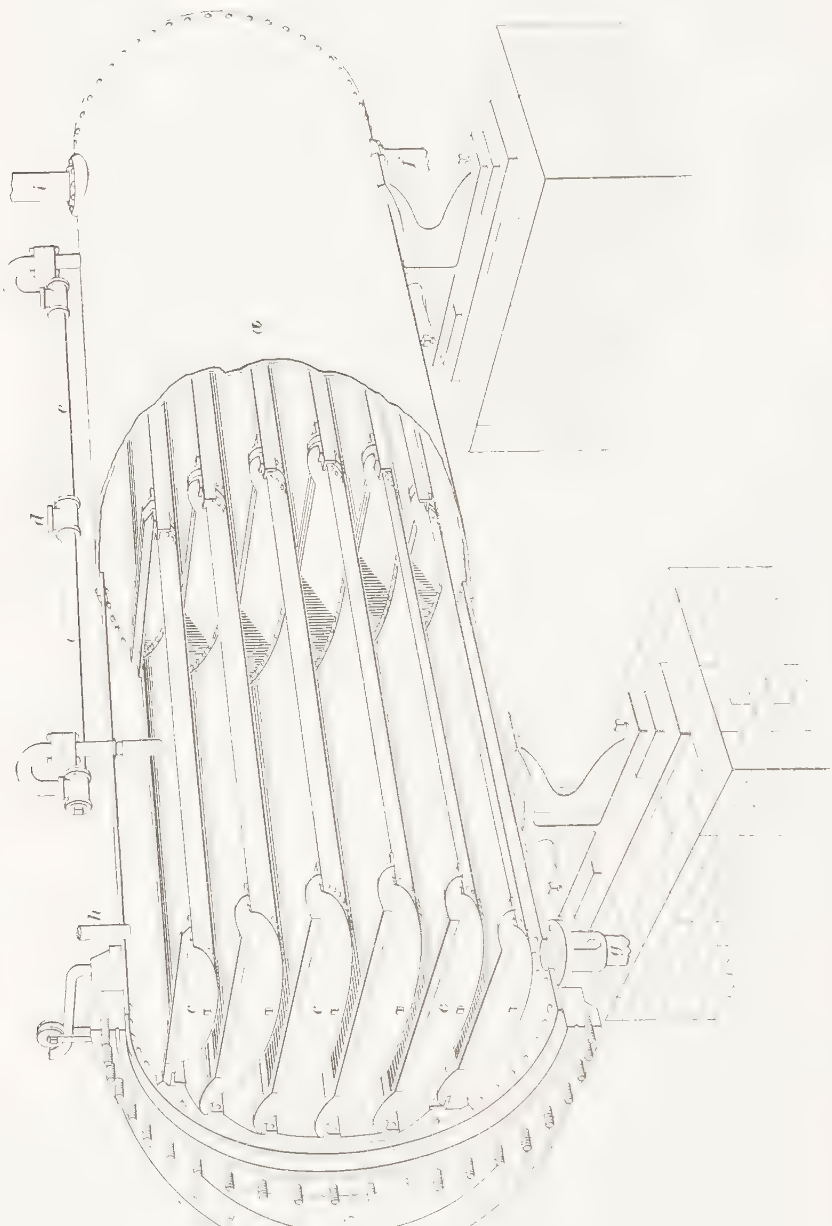


Fig. 14.

90. Live-steam feedwater heaters are made in a variety of ways; some are constructed with removable pans, others depend on the use of a blow-off and occasional opening up of the heater for the removal of the sediment.

91. The **Hoppes purifier** shown in Fig. 14 belongs to the removable-pan type. It consists of a cylindrical shell *a* fitted on one end with a removable head *b*, shown in the figure as taken off and swung out of the way. A series of shallow steel pans *c*, *c* are placed within the purifier. The feedwater enters the **T** shown at *d* and flows through the branch pipes *e*, *e* into the top pans. The feedwater flows in thin sheets over the edges of the pans and finally out of the purifier through the pipe *f*. A glass water gauge, not shown in the figure, connects the blow-off pipe *g* with the nipple *h* and shows the amount of water. Live steam from the boiler enters through the pipe *i* and heats the water in the purifier to a temperature nearly equal to its own, and in so doing precipitates those impurities that become insoluble by heat. Mud and earthy matter deposit on the inside of the pans, while the scale-forming substances coat the outside of the pans. The feedwater flows through *f* into the boiler by gravity, the purifier being placed higher than the boiler and having a pressure equal to that in the boiler within it.

92. The sediment must be removed periodically, the pans being easily removable for the purpose. A purifier of this kind will quite effectually remove the impurities from the feedwater when used with steam pressures of over 50 pounds. With lower pressures the temperature is not sufficient to precipitate all the scale-forming substances.

93. A **Buffalo feedwater heater and purifier,** which can be applied to any boiler, is shown in Fig. 15. The feed-pumps deliver their water through the pipe *e* and check-valve *f* into the top of the heater. The entering feedwater strikes against the top division plate; the solid stream of water is thus broken up. It now flows in a zigzag course over the edges of the spray disks *g*, *g*, being thus spread out into large, thin sheets that readily absorb the heat of the live

steam with which the spray chamber is filled and which is admitted through *d*. The highly heated water falls to the bottom, and passing around the division plate *l* and deflector plate *m*, rises in the settling chamber *a*. Thence it passes through the equalizing tube *i* into the space between the upper division plate and head and into the feedpipe *c*.

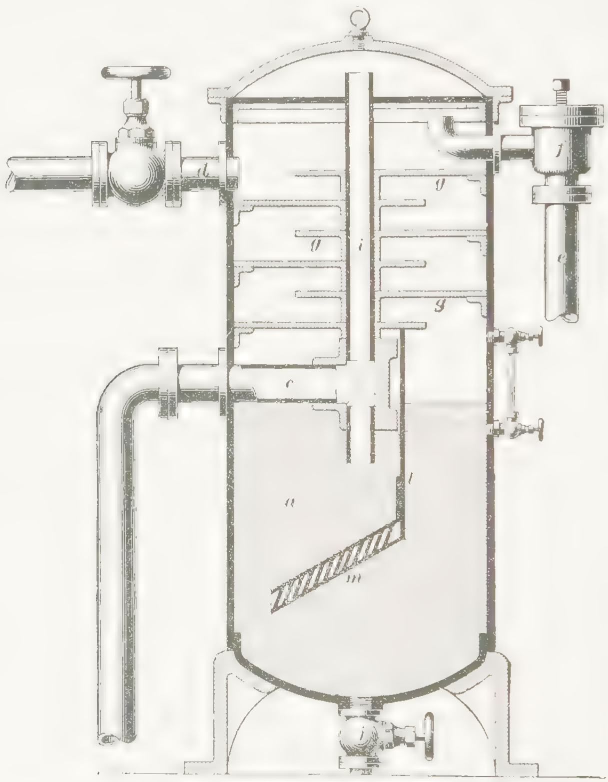


FIG. 15.

The feedwater being heated to almost the same temperature as that in the boiler, the scale-forming substances precipitate and collect in the bottom of the settling chamber, whence they can be removed by opening the blow-off valve *j*. Nearly all foreign matter in mechanical suspension also collects in this settling chamber and is removed with the

scale-forming substances. The impurities that have a smaller specific gravity than water rise to the top of the settling chamber and float on the water. By extending the equalizing pipe *i*, which forms the feed outlet from the heater, below the surface of the water, the floating impurities are kept from entering the feed outlet. The heater should be placed above the boilers. The water will then flow into the boilers by gravity. To prevent the water in the boilers backing up into the heater when the blow-off of the heater is opened, an automatic shut-off valve, which is simply a special form of check-valve, is supplied. This valve is placed in the feedpipe between the heater and the boilers. Under ordinary working conditions, it is sufficient to blow out the heater once every six hours.

TESTING WATER.

94. Introduction.—A quantitative analysis cannot be made by any one except an expert chemist having a well-appointed laboratory and the proper apparatus; a qualitative analysis for the most common impurities can be easily made, however, with the aid of chemicals procurable in almost any drug store. Such an analysis will show what kind of impurities are present, but will not show the amount. The tests for the more common impurities are given below.

95. Testing for Corrosiveness.—It is a good plan to test the feedwater and also the water in the boiler occasionally for corrosiveness. This may be done by placing a small quantity into a glass tumbler and adding a few drops of methyl orange. If the sample of water is acid, and hence corrosive, it will turn pink. If it is alkaline, and hence harmless, it will be yellow. The acidity may also be tested by dipping a strip of blue litmus paper into the water. If it turns red, the water is acid. This method is not as sensitive as the previous one, which should be used in preference. If litmus paper is kept in stock, it should be kept in a bottle with a glass stopper, as exposure to the atmosphere will deteriorate the paper. If the water in the boilers has

become corrosive and corrosion has set in, the water in the gauge glass will show red or even black. As soon as the color is beyond a dirty gray or straw color, it is advisable to introduce lime or soda to neutralize the acid

96. Testing for Carbonate of Lime.—Pour some of the water to be tested into an ordinary tumbler. Add a little ammonia and ammonium oxalate; then heat to the boiling point. If carbonate of lime is present, a precipitate will be formed.

97. Testing for Sulphate of Lime.—Pour some of the feedwater into a tumbler and add a few drops of hydrochloric acid. Add a small quantity of a solution of barium chloride and slowly heat the mixture. If a white precipitate is formed, which will not redissolve when a little nitric acid is added, sulphate of lime is present.

98. Testing for Organic Matter.—Add a few drops of pure sulphuric acid to the sample of water. To this add enough of a pink-colored solution of potassium permanganate to make the whole mixture a faint rose color. If the solution retains its color after standing a few hours, no organic substances are present.

99. Testing for Matter in Mechanical Suspension. Keep a tumblerful of the feedwater in a quiet place. If no sediment is formed in the bottom of the tumbler after standing for a day, there is no mechanically suspended matter in the water.

FEEDWATER HEATERS.

100. Purpose.—It is important that the feedwater should be introduced into the boiler at as high a temperature as possible. The advantages of hot feedwater are (1) The avoidance of the strains produced in the plates of the boiler by the introduction of cold feedwater. (2) The saving in fuel effected by the higher temperature of the feedwater. In order that there be a direct saving in fuel, it is necessary that the heat used for heating the feedwater

be taken from a source of waste, the principal ones being the waste gases and the exhaust steam. Only exhaust-steam feedwater heaters will be considered here, since live-steam feedwater heaters have already been treated of in Arts. **89** to **93**.

101. Economy.—The economy of using hot feedwater may be shown by a simple calculation. Suppose a boiler to furnish steam at 75 pounds pressure and let the feedwater temperature be 60° F. The number of heat units required to change a pound of water at 60° into steam at 75 pounds pressure is, from the Steam Tables, about 1,151 B. T. U. Suppose, in the second case, that the feedwater enters at a temperature of 210°. Then, the number of heat units gained by heating the feedwater is 210 − 60 = 150 B. T. U., and the gain, in per cent., is $\frac{150 \times 100}{1,151} = 13$ per cent.

102. Classification.—Exhaust-steam feedwater heaters are divided into two general classes, known as *open heaters* and *closed heaters*.

103. An **open heater** may be defined as a heater in which the water is in contact with the atmosphere. Open heaters may be divided into two subclasses, known as **direct-contact open heaters** and **coil heaters**. In a direct-contact open heater the exhaust steam comes into contact with the water, which, by means of suitable devices, is broken into spray or thin sheets in order to readily absorb the heat of the steam. In a coil heater the exhaust steam passes through coils of pipe submerged in a suitable vessel containing the water to be heated and open on top. Such heaters are generally improvised affairs made directly on the premises.

104. Closed exhaust-steam feedwater heaters may be defined as heaters in which the feedwater is not exposed to the atmosphere, but is subjected to the full boiler pressure.

The steam does not come into contact with the water; the latter is heated through coming into contact with metallic surfaces, generally those of tubes, that are heated by the exhaust steam.

OPEN HEATERS.

105. Comparison and General Considerations.—In an open direct-contact heater the exhaust steam and feedwater mix, most of the steam being condensed. Consequently, all the oil or grease carried over from the engine in the exhaust steam will mingle with the heated water, and in order to prevent this oil entering the boiler, special provision must be made for intercepting it, either by filtration or by means of some skimming apparatus. In a coil heater the steam and water do not mingle, and, consequently, no provision for the removal of oil is necessary. Open heaters must always be placed on the suction side of the feed-pump, and since they furnish hot water to the pump, it is necessary to place them several feet above the latter in order that the water may flow into it by gravity. Direct-contact open heaters will, in general, heat the feedwater to a higher temperature than open heaters of the coil type and cause less back pressure on the engine. With a direct-contact open heater a temperature of 210° is often obtained, and at this temperature most of the scale-forming substances, with the exception of the sulphates, will precipitate, so that such a heater incidentally acts as a purifier. In addition, most of the matter in mechanical suspension settles to the bottom. For the removal of sulphates a live-steam purifier must be used. A coil heater, in general, will not heat the water sufficiently hot to make any scale-forming substances insoluble; it will, however, heat the water hot enough to cause a marked economy in the use of fuel. Open heaters should never be used with an injector, since the injector requires the water flowing to it to be cold in order to work.

106. Examples of Open Direct-Contact Heaters.—Fig. 16 shows the construction of a very simple and quite effective home-made open heater, which was designed by Mr. F. M. Humason and is used by him. The heater is intended for a 12″ × 24″ Corliss engine making 95 revolutions per minute. The body of the tank *A* is made of 2″ × 4″ pine, well bound together with 5 iron hoops and fitted with strong wooden heads; a manhole *B* is placed in the top head. The tank is 3 feet in diameter by 5 feet 6 inches in height. About 2 feet 6 inches from the top is placed a grating *C* made of 2″ × 4″ pine with the bars 1 inch apart; this grating is covered with burlap and 12 inches of hay. The exhaust steam from the engine enters at *D* and meets the cold feedwater coming through the pipe *E*, the water supply in this case coming from the tank *F*. Any uncondensed exhaust steam passes out of the pipe *G*. The heater is fitted with a by-pass, so that it may be cut out of service. This is done by opening the valve in *I* and closing the valves in *E* and in the suction pipe *H*, when the water in the tank *F* will pass through *I* into the suction pipe. A gauge glass *K* shows the depth of water in the heater. The exhaust from the pump enters the heater through the pipe *L*. The feedpipe is tapped for the pipe of the small soda tank *M*, which affords a ready means of introducing soda into the feedwater. The hay and burlap require to be renewed quite frequently.

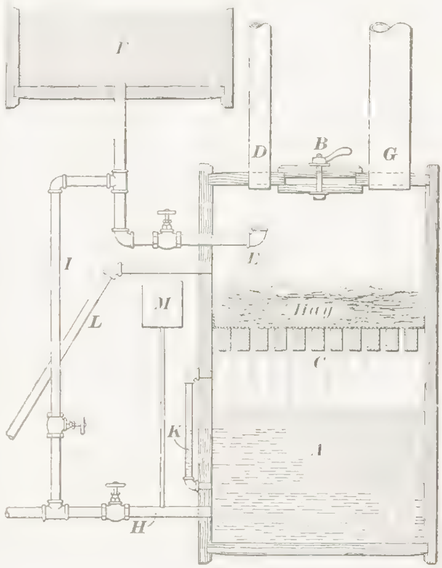


FIG. 16

107. The **Pittsburgh open feedwater heater** is shown in section in Fig. 17. The cold water is admitted to the heater through a balanced valve *a*. It then passes through the inlet pipe in the center of the upper head to a spraying device *b*, located immediately above the pans. In passing through this spraying device, the water is distributed evenly in a very fine spray to the upper pan *c*. From the inner edge of the upper pan it falls vertically to the pan below, and from this pan over the outer edge to the one beneath, and so on through the whole series. The pans being arranged so as to give the water a zigzag travel from the one to the other, two fine cylindrical sheets of water are maintained, one each on the inner and outer edges of the pans.

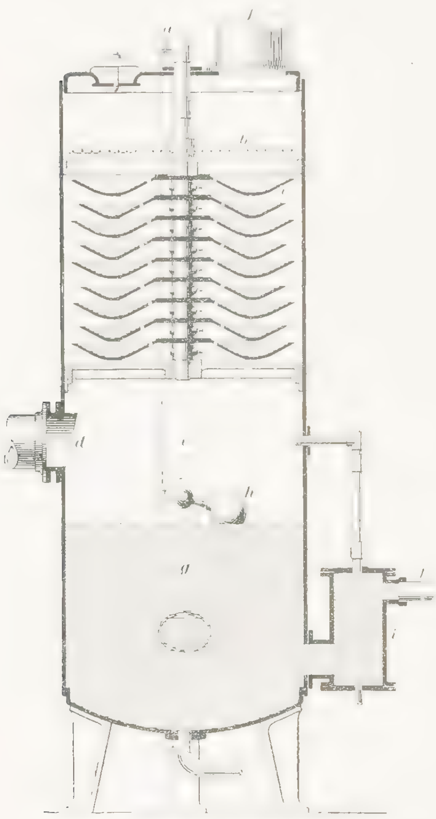


Fig. 17

108. The exhaust steam enters at *d*, immediately below the distributing pans, and passes into a large steam space *e*, from which it rises slowly, coming into direct contact with the fine sheets of falling water. Some of the steam is condensed by coming into contact with the cool water; the remainder passes into the atmosphere through the outlet

connection *f* on top of the heater. The feedwater, being spread into large thin sheets, on meeting the exhaust steam readily absorbs its heat until the highest temperature attainable (212° F.) is reached. The purification of the feedwater also takes place at this point in the heater, the water depositing such impurities as can be precipitated at a temperature of 212° F. in the pans. After leaving the pans the water falls through the steam space to the water reservoir and sedimentation chamber *g* in the lower part of the shell. The volume of this reservoir is made large to provide a quiet place for the water, thus permitting a free and undisturbed settling of the foreign matter held in mechanical suspension in the water coming from the pans. Some users insert hay, etc. in this reservoir to help purify the water, but this is deemed unnecessary by the makers.

109. The water level is maintained in this reservoir by means of a float *h*, which operates the balance valve *a* of the cold-water supply pipe on the top of the heater through two cranks and a connecting-rod. Should the water become low, the float falls and opens the balance valve, thus admitting more water. In the event of too much water the float rises and shuts off the valve. An automatic regulation of the water supply is by this means maintained at all times, requiring no attention from the man in charge beyond a casual look at the gauge glass placed at the water-level on the outside of the heater. As previously mentioned, a further purification of the water takes place in this reservoir, the mud and heavier impurities settling to the bottom of the shell, where they can be blown off.

110. The oil and lighter impurities float on top of the water and are taken care of by a skimming apparatus in the shape of a small cylinder *i* placed on the outside of the shell and connected to the water reservoir by a large flanged connection at the point shown. It is also connected on top by a pipe to the steam space of the heater. The feed-pump takes its water supply through the suction pipe *k*

attached to the upper part of this small cylinder. The object of connecting the top of the separator *i* with the steam space of the shell is to prevent the water level in the shell falling to any extent below the level of the pump connection. As soon as it falls slightly below this level, steam from the steam space of the heater will flow into the top of the separator and thence into the suction end of the pump, thus preventing the latter taking any more water from the heater. It naturally follows from this construction that the oil and other floating impurities can never enter the separator and consequently cannot be pumped into the boiler.

---

111. Example of a Coil Open Heater.—Fig. 18 shows the general construction of a coil heater using exhaust steam. The water to be heated is contained in an iron or wooden tank into which it is discharged through the supply pipe *d* and which it leaves through the pump suction pipe *e*. The exhaust steam from the engine enters through the pipe *b* and after passing through the coil *a* leaves through the pipe *c*. The coil should be so arranged

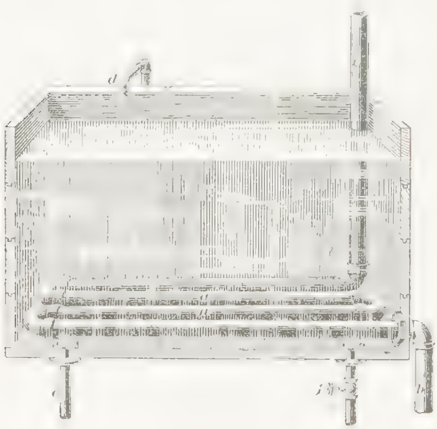


FIG. 18

that the end farthest from the end where the exhaust steam enters is lowest, and a drain pipe leading to a steam trap should be fitted there in order that any condensed steam may be drained automatically from the coil. The tank holding the water may be fitted with a light removable cover, in order to keep dirt out of the water, and should be fitted with a glass water gauge, or an equivalent device,

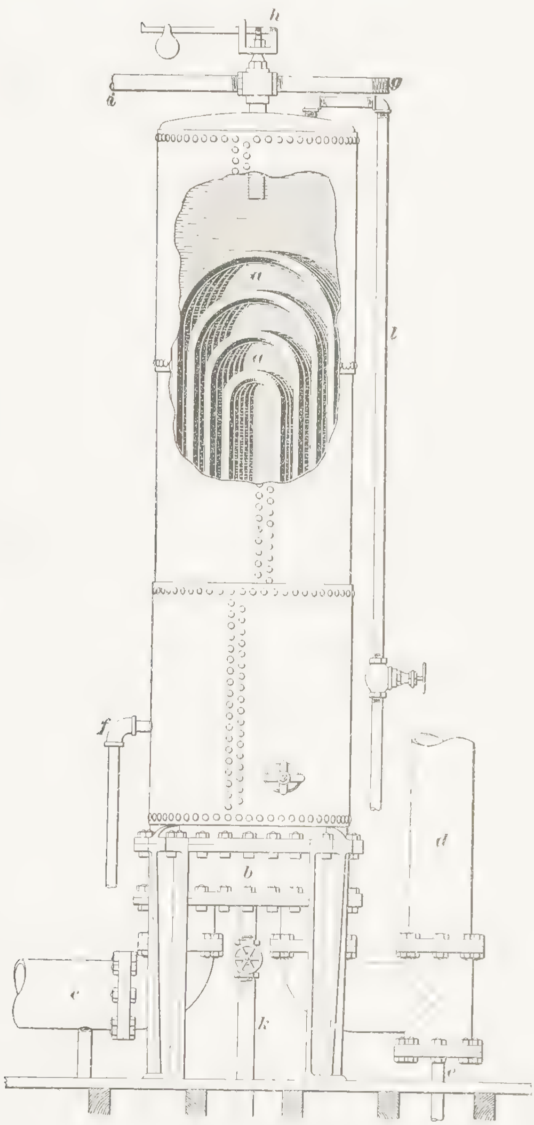


FIG. 19.

and a blow-off, as *f* in the figure, for emptying it. About $\frac{1}{10}$ square foot of heating surface may be allowed in the coil for every pound of water to be heated per hour. Thus, to heat 3,000 pounds of water per hour requires $3,000 \times \frac{1}{10} = 300$ square feet of heating surface in the coil.

CLOSED HEATERS.

112. In a closed exhaust-steam heater, the steam does not come into contact with the water, and the heater is placed between the feed-pump, or injector, and the boiler, so that the feed-pump handles cold water. The heating surface is generally made of tubes; sometimes the water is inside the tubes and sometimes it is outside, the latter being the more general plan. Closed heaters are made in a great variety of forms, all of which, however, embody the same general principles. For this reason only one is shown here.

113. Fig. 19 shows the **Berryman closed feedwater heater.** The heating surface is obtained by means of inverted **U** tubes *a*, *a* through which the exhaust steam passes. The steam enters the bottom *b* of the heater through the pipe *c*, and there being a steam-tight partition in the bottom, is caused to flow up through one leg of the tubes and down the other and then through the pipe *d* to the atmosphere. The condensed steam is discharged through the drip *e*. The feedwater enters the heater through the pipe *f* and leaves at the top through the feed-pipe *g*. A safety valve *h* is fitted to the heater, which prevents any overpressure in it due to a closing of the globe valve in the feedpipe before the pump is stopped. When the safety valve is open, the water discharges through *i* to the sewer. The heater is fitted with a bottom blow-off *k* and a surface blow-off *l* for removing floating and settled impurities in the water. It heats the water to a temperature between 200° and 212°, and in so doing precipitates those scale-forming substances that become insoluble at the temperatures given.

PUMPS.

(PART 1.)

GENERAL INTRODUCTION.

DEFINITION.

1. Pumps are machines for lifting or conveying fluids, and when not otherwise specified the word is generally understood to mean machines for lifting and conveying water.

WATER.

2. Water is a liquid composed of 1 part of oxygen and 2 parts of hydrogen. The weight of a cubic foot at its maximum density (39.2° F.) is 62.425 pounds; at 32° F., or the freezing point, water weighs 62.4 pounds per cubic foot, and at 212° F., or the boiling point, water weighs 59.7 pounds per cubic foot. Obviously pumping machinery can only handle water between the limits of the freezing and boiling points. Water is almost non-compressible; its compressibility is about .0000014 of its volume under a pressure of 15 pounds per square inch, and it decreases with an increase of temperature; for practical purposes it may be considered as incompressible.

HOW WATER FLOWS INTO A PUMP.

3. Pumps are frequently so located that the water must flow into the pump cylinder by atmospheric pressure on the surface of the water external to the suction pipe; that is, by

the action of the pump a vacuum of more or less perfection is produced in the pump chamber. If the end of the suction pipe, which is the pipe connecting the pump chamber with the water, is submerged, the excess of pressure on the surface of the water outside of the suction pipe will cause the water to rise in the suction pipe until the pressure due to the weight of the column equals the pressure of the atmosphere.

THEORETICAL LIFT.

4. The pressure of the atmosphere is constantly changing. For practical purposes the pressure at sea level is taken as 30 inches of mercury, or 14.7 pounds pressure per square inch. Since a pressure of 1 pound per square inch is equal to that exerted by a column of water 2.309 feet high, the theoretical height that water can be raised by a perfect vacuum at sea level will be $14.7 \times 2.309 = 33.94$ feet. Since the atmospheric pressure becomes less as the altitude increases, it follows that the greater the altitude, the less the theoretical and practical lift by atmospheric pressure will be. To find the theoretical height in feet to which water can be lifted at any altitude, multiply the barometric reading in inches by 1.133.

5. For water holding foreign substances in suspension, or for other liquids, the theoretical lift can be found by dividing the theoretical height to which water can be lifted at the existing atmospheric pressure, as shown by the barometer, by the specific gravity of the liquid.

ACTUAL LIFT.

6. Since a perfect vacuum cannot be obtained on account of mechanical imperfections, air contained in the water, and the vapor of the water itself, the actual height to which it can be lifted is only about .82 of the theoretical height, which ratio is good only for pure water.

7. In the case of pumps located at the bottom of deep mines, the barometer will plainly show a greater pressure on

the bottom than at the surface, and hence a greater suction lift is possible at the bottom.

PUMPING HOT WATER.

8. Pumping hot water is a difficult problem and has positive limitations in the direction of lift and temperature. Whenever possible, the pump should be so arranged that the hot water will flow to it. The following table shows the theoretical possibilities at 30 inches barometric pressure:

INFLUENCE OF TEMPERATURE ON SUCTION LIFT.

Temperature. Degrees Fahrenheit.	Suction Lift. Feet.
100	28.0
150	21.0
170	17.0
190	10.0
210	1.5

In actual practice it is not possible to lift water at all whose temperature exceeds 180°. The reason hot water cannot be lifted is on account of the increased pressure of the vapor at the higher temperatures. Pumps required to handle hot water should be provided with suitable valves of vulcanized rubber for the lower temperatures and metal for the higher temperatures. Soft rubber valves are unsuited for handling hot water.

LIMIT OF HEIGHT TO FORCING.

9. Having considered the limit of lift by suction, the limit of height to which water or any other liquid can be forced will be discussed now. This height is *not* affected by

the atmospheric pressure and is only limited by the power available for forcing the liquid and the strength of the pump and the pipe connections.

GENERAL CONSIDERATIONS.

10. Before proceeding to give an account of some of the best and most modern types of pumps, let us consider for a moment what is required to be done in order that large volumes of water may be raised in the best possible manner and with the best possible economy. Among the first things that a practical engineer should know and among the last things he should forget is that in handling water within pipes he has a fluid which, while it is flexible to the greatest extent and is susceptible to the influence of power or force of greater or less intensity, and while it may be drawn from below and raised to a height above, and while it bends itself to the will of the engineer, will still refuse to do some things and which all the complicated appliances of the engineer have as yet failed to compel it to do. When enclosed within chambers and pipes to an extent that fills them, it will not permit the introduction of any more without bursting them. When enclosed within long lines of pipes, it will *not* suddenly start into motion or when in motion suddenly come to rest without shocks or strains more or less disastrous.

HISTORICAL.

11. Almost the first application of steam was to pumps used for raising water out of mines, and as these pumps had previously been entirely operated by horses, a basis of comparison was established by rating the power of the steam engine by the number of horses it displaced at these mine pumps. To enumerate even briefly the various machines for pumping water that have been developed in the past, many of which are famous, would be quite impossible for lack of space, and a description of their peculiar and prominent characteristics would be equally so, especially as they are only of historic interest. Conditions have so changed

as regards steam pressure, speed, and problems of manufacture and competition that out of the great mass of pump designs, some of which were excellent in many points, have been developed standard forms of pumps particularly adapted to each and every service.

CLASSIFICATION.

12. Pumps may be classified in a number of different ways, as according to their principle of operation, their general form, the power used to drive them, the methods of applying the power, the special class of work to which they are applied, etc. There being no universally accepted classification, no attempt will be made here to classify pumps, but the different forms of pumps described will be given the name by which they are most generally known.

STEAM PUMPS.

DEFINITION AND DIVISION.

13. Steam pumps are pumps in which the moving force is steam, which is applied to the movement of water without the intervention of belting or gear-wheels. Steam pumps are divided into two general classes known, respectively, as *direct-acting* and *flywheel-pattern* pumps.

DIRECT-ACTING STEAM PUMPS.

INTRODUCTION.

14. General Description.—The type of pump by far the most numerous is the **direct-acting pump,** by which is meant a steam-driven pump in which there are no revolving parts, such as shafts, cranks, and flywheels, or pumps in which the pressure of the steam in the steam cylinder is

transferred to the piston or plunger in the pump in a direct line and through the use of a continuous rod or connection. In pumps of this construction the moving parts have no weight greater than that required to produce sufficient strength in such parts for the work they are expected to perform; as there is, consequently, no opportunity to store up power in one part of the stroke to be given out at another, it is impossible to cut off steam in the cylinder during any part of the stroke. The uniform and steady action of the direct-acting steam pump is dependent alone on the use of a steady, uniform pressure of steam throughout the entire stroke of the piston against a steady, uniform resistance of water pressure in the pump, the difference between the force exerted in the steam cylinders over the resistance in the pump governing the rate of speed at which the piston or plunger of the pump will move. The length of the stroke of the steam piston within these pumps is limited and controlled by the admission, release, and compression of the steam in the cylinder.

15. Development.—The direct-acting steam pump was invented by Henry R. Worthington in the year 1840 and was patented in 1841. A few years later Mr. Worthington developed and brought out what is now known as the *duplex* direct-acting steam pump. The objection to the single direct-acting pump was the fact that the action of the pump plunger or piston was an intermittent one; that is, the column of water was started into motion at the beginning of each stroke and came to a stand at the end of each stroke, thus not only making the flow of the water irregular, but also subjecting the pump and the connecting pipes and their joints to severe and often serious strains.

16. Duplex Pumps.—In the main, the construction of the steam and water ends of the duplex pump differs but slightly from that of the single direct-acting pump, but the mechanism that operates the steam valves is different and the effect on the water column is very different. The principle upon which the duplex pump operates is this: Two

pumps of similar construction are placed side by side; a lever attached to the piston rod of each pump connects to the slide valve of the opposite steam cylinder, and thus the movement of each steam piston, instead of operating its own steam valve, as in the single pump, operates the slide valve of the opposite cylinder. The effect of this arrangement is that as the piston or plunger of one pump arrives at or near the end of its stroke, the plunger or piston of the other begins its movement, thus alternately taking up the load of the water column and producing a regular, steady, onward flow of water without the unusual strains induced by such a column of water when suddenly arrested or started in motion.

17. Advantages of Direct-Acting Pumps.—The direct-acting machine is the simplest form of pump yet devised; its action most nearly harmonizes with the laws controlling the action of water, and in event of a conflict, the direct-acting pump will yield to the superior force of the water without serious resistance. The direct-acting pump being not only the simplest but most universally used type of pump it will be taken up first.

18. Disadvantages of Direct-Acting Pumps.—To obtain perfection of steam pumps it is necessary to use the steam so that by cutting it off within the steam cylinders and by subsequent expansion in the same or other cylinders, its expansive force will be developed to the highest limit and to the most economical extent. When that is done we have accomplished all that with our present knowledge of the steam engine can be done in the steam cylinders. It is in this respect, however, that the direct-acting steam pump of the ordinary type is anything but economical, its design requiring the carrying of the full steam pressure throughout the whole stroke. This drawback prohibits its use in places where a high economy in the use of steam is imperative. By the use of a so-called **high-duty attachment,** however, the ordinary direct-acting pump can be and is converted into a machine using steam expansively; a fair degree of economy is also obtained by compounding the steam end.

VALVE MOTIONS.

19. The Knowles Valve Motion.—Fig. 1 shows the steam end of the Knowles steam pump with the arrangement of the valve gear.

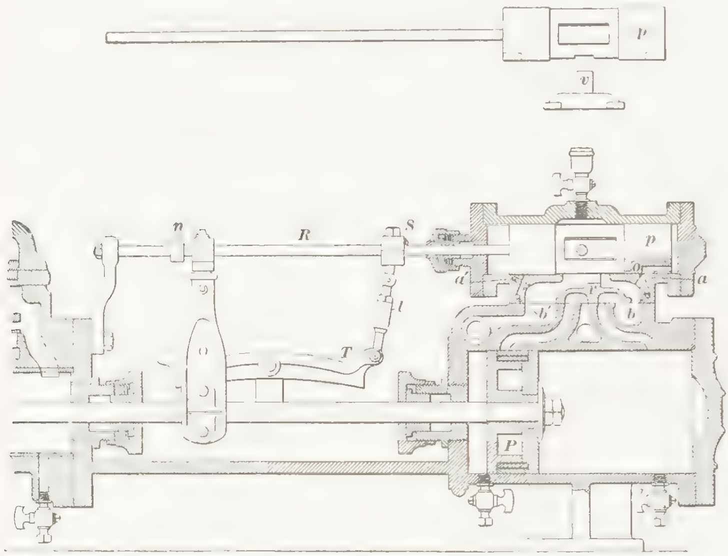


FIG. 1.

An auxiliary piston *p* works in the steam chest and drives the main valve *v*. This auxiliary, or **chest piston,** as it is called, is driven backwards and forwards by the pressure of the steam, carrying with it the main valve, which in turn gives steam to the steam piston *P* and operates the pump. The main valve *v* is a plain slide valve of the **B** form working on a flat seat. The chest piston has a rod *R* to which is clamped an arm *S*, the latter being connected to the rocker bar *T* by a link *l*. The main piston rod carries an arm *O*, which is provided with a stud, or bolt, on which there is a friction roller. This roller moves back and forth under the curved rocker bar with the motion of the main piston rod and lifts the ends of the bar, thus giving the chest piston a slight rotary motion just at the end of the stroke of the main piston.

Each end of the chest piston is provided with a port *o*, shown in the right-hand end by the partial section, and the solid part of the steam chest has four ports *a*, *b* and *a'*, *b'*, which open into the space in which the chest piston moves. The ports *a* and *a'* connect with the live steam space in the steam chest and serve as steam ports, while *b* and *b'* connect with the exhaust. In the position shown in the figure, the main piston has just reached that point of its stroke where the roller has acted on the rocker bar to rotate the chest piston. This has brought the port *o* (in the right-hand end of the chest piston) into communication with the live steam, admitting the latter to the space at the right of the chest piston. This steam drives the chest piston to the left and it carries the main valve *v* with it, thus exhausting the steam from the right of the main piston and admitting live steam to the left. When the main piston, under the action of this steam, approaches the right end of the cylinder, the roller lifts the right end of the rocker bar, thus rotating the chest piston so as to bring the port *o* in connection with the exhaust port *b* and the port in the opposite end of the chest piston in connection with the steam port *a'*. This drives the chest piston and main valve to the right, allows the steam at the left of the main piston to exhaust, and admits live steam to the right of the main piston again. The chest piston, as it approaches either end of its chamber, covers the exhaust port at that end, thus confining enough of the exhaust steam to form a **cushion** to prevent it from striking the end of the steam chest. The main piston also covers the exhaust port before reaching the end of its stroke, as shown in the figure, so that it is cushioned by the exhaust and prevented from striking the cylinder head. Special passages are provided for admitting the steam required to move the piston far enough to uncover the main ports on the return stroke. The arm *O* carries a collar that slides over the chest piston rod, and in case the steam pressure is not sufficient to move the chest piston, this collar will strike collars, as *n*, and thus move the valve. (One of these collars is just behind the arm *S*.)

20. The Cameron Valve Motion.—In the Cameron pump shown in Fig. 2, which possesses the advantage of

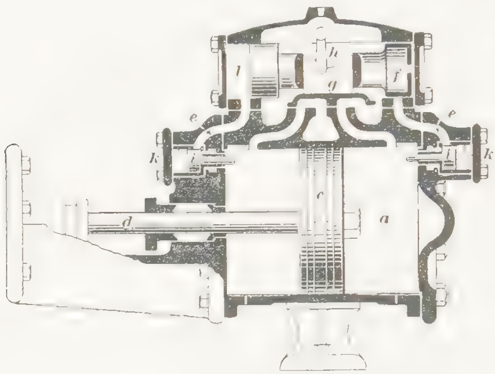


FIG. 2.

having no outside gearing, *a* is the steam cylinder, *c* the piston, *d* the piston rod, *l* the steam chest, *f* the chest piston, the right-hand end of which is shown in section, *g* the main slide valve, *h* the starting bar, connected with a handle on the outside, *i*, *i* the reversing valves, *k*, *k* the bonnets over the reversing-valve chambers, and *e*, *e* are exhaust ports leading from the ends of the steam chest direct to the main exhaust and are closed by the reversing valves *i*, *i*.

The action of this valve motion is as follows: The spaces at the ends of the chest piston *f* communicate with the live-steam space by means of small holes, one of which is shown in the right-hand section of *f*. By means of these holes, these spaces and the ports *e*, *e* leading from them are kept filled with live steam as long as the ports are covered by the piston valves *i*, *i*. In the position shown in the figure, the space in the main cylinder to the right of the piston *c* is in communication with the live-steam space in the steam chest; *c* is therefore moving to the left. When *c* strikes the stem attached to the valve *i*, it forces *i* to the left and uncovers the left-hand port *e*, thus allowing the steam at the left of *f* to pass out through the exhaust. The steam to the right of *f* then expands and drives *f* and with it the main

valve *g* to the left, thus reversing the action of the steam on *c*, which immediately begins to move back towards the right. Live steam is always acting on the piston *i*, so that as soon as *c* moves to the right, this steam pushes *i* back and covers the left port *e* again, after which live steam fills the port and the space connecting with it through the small hole in the end of *f*. When the piston *c* strikes the stem of the right-hand valve *i*, the main valve is again shifted to the right and *c* is started on its stroke to the left. Exhaust steam is confined in the ends of the cylinder to prevent the piston from striking the heads, in the same manner as in the Knowles steam pump.

21. The Gordon Steam Pump.—If the load is suddenly thrown off from the ordinary direct-acting steam pump

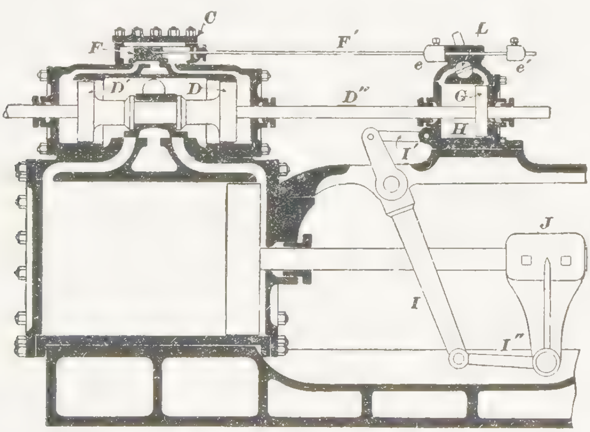


FIG. 3.

through any cause, as, for example, the bursting of the discharge pipe or the opening of a valve, so as to permit the water to discharge freely under low pressure, the steam is liable to drive the piston to the end of its stroke with so much force as to cause serious shocks or even to break some part of the pump. In order to overcome this danger the *Gordon* steam pump is provided with the arrangement shown in Fig. 3, which is called an **isochronal valve gear.**

In this gear the main valve is operated by a double chest piston $D D'$, which is actuated by steam controlled by an auxiliary slide valve F in the small steam chest C. F is provided with a valve stem F', to which two collars e, e' are fastened with setscrews. A slide H, which receives its motion from the main piston rod by means of links I' and I'', the lever I, and the crosshead J, strikes the collars e, e' near the ends of the main piston stroke, thus moving the auxiliary valve F and admitting steam to the chest piston $D D'$, which in its turn operates the main steam valve and reverses the motion of the main piston. In order to prevent the pump from running away when the load is thrown off suddenly, the slide H carries a cylinder in which works a piston G fastened to the rod D'' of the chest piston $D D'$. This cylinder, called the **cataract cylinder,** has a cock L that controls a passage joining its two ends, and by means of this cock the passage may be more or less closed, as desired.

The action is as follows: Assume the cataract cylinder to be empty; the piston G will then meet with no resistance and the machine will work as usual. At the end of the stroke the slide H will strike one of the stops e or e', thus shifting the auxiliary valve F and admitting steam to the piston valve $D D'$, which will move freely through its stroke and thus admit steam to the main piston for the return stroke. Now, if something happens to the water discharge, as, for example, the breaking of a pipe, the load will be removed from the pump and the main piston will be driven suddenly to the end of its stroke and thus be in danger of striking the head with enough force to break it. The object of the cataract cylinder is to overcome this danger. It is filled with liquid, which must be forced from one end to the other by the motion of the piston G. By partly closing the cock L a resistance is opposed to the passage of the liquid, and the motion of the piston G through its cylinder may be made as slow as desired; consequently, when the main piston moves too rapidly, the motion of the slide H will be transmitted to the piston G, which will shift the main valve

so as to shut off the supply of steam to the main piston and thus prevent the pump from running too fast.

22. The Marsh Steam Pump.—The **Marsh valve motion** shown in Fig. 4 operates without any connection to

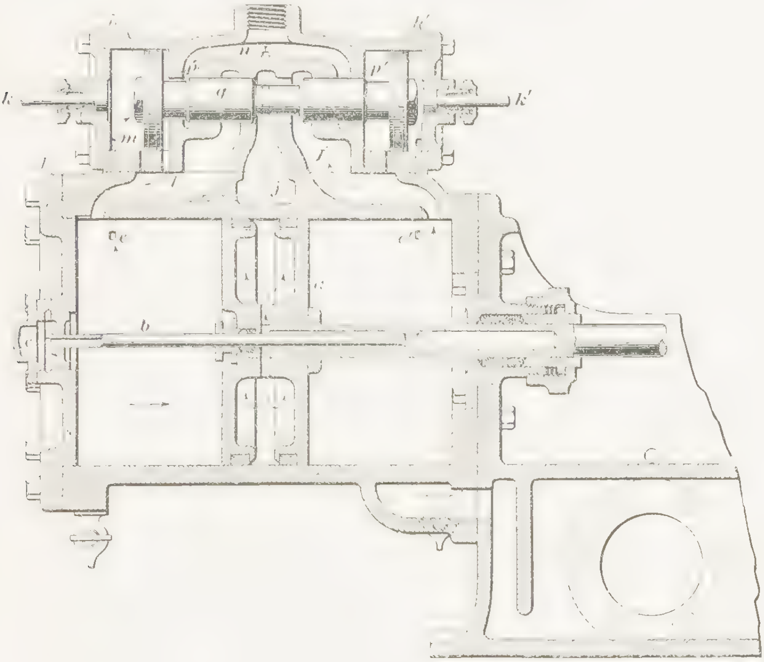


FIG. 4

the piston or rod. The steam piston *a* is made in two parts, each section being provided with a packing ring so arranged as to provide an annular space between the two piston halves. Steam at boiler pressure is admitted within the pistons by means of the tube *b*, which is rigidly secured to and is in communication with the chamber *c*. A stuffingbox in the piston through which it plays prevents leakage into the main steam cylinder. A small port *d*, shown by dotted lines, supplies steam to the chamber *c*. When the piston is moving to the right, steam is entering from the space *n* through the annular opening between the reduced neck of the valve *g* and

bore of the left chest wall *p*. It is thus projected against the inside surface of the valve head *h* before escaping through the port and passing into the cylinder. Both the pressure and the impulse due to the velocity of the entering steam act on the valve head *h* and tend to force it to the left, thus tending to close the annular opening in the chest wall *p*. The steam flowing through the annular opening and port *l* into the cylinder also flows through the small ports *m* and *e* to the left of the valve head *h*. The steam entering through these ports is wiredrawn, so that its pressure is reduced, but it has a greater area of the valve head *h* exposed to its pressure than the steam on the right of *h*. Hence, the valve *g* moves to a position where the total forces acting on the two sides of *h* are equal and then remains stationary. The steam entering through the annular opening in the chest wall *p* is also wiredrawn, so that the pressure on the left of the piston *a* is below the full boiler pressure existing in *n*.

While the piston *a* is moving to the right, the steam on the right is exhausting through the port *f* into the exhaust port *j*. The exhaust is first closed by the piston running over the port *f*; as soon as this port is covered, the port *e'* leading to the right of the valve head *h'* communicates with the space within the piston containing steam at boiler pressure, and this live steam rushes into the space to the right of the valve head *h'*. Since the steam pressure on the left of *h* is less than the pressure on the right of *h'*, the valve moves to the left, and by doing so closes the left steam inlet, opens the left exhaust, and also opens the right steam inlet in the chest wall *p'*. The live steam admitted to the right of the piston *a* first brings it to rest and then reverses its motion. The tappets *k* and *k'* are used for moving the valve by hand in case the valve is stuck.

DUPLEX PUMPS.

23. Types.—Duplex pumps, like single direct-acting pumps, are made either as piston pumps or as plunger pumps. When made as plunger pumps, they may have either

inside-packed, center-packed, or outside-packed plungers. Piston pumps are preferred for moderate pressures, but for pumping against very high pressures the plunger pump is generally used.

24. Slide-Valve Worthington Duplex Pump.—Fig. 5 is a perspective view of a piston pattern Worthington

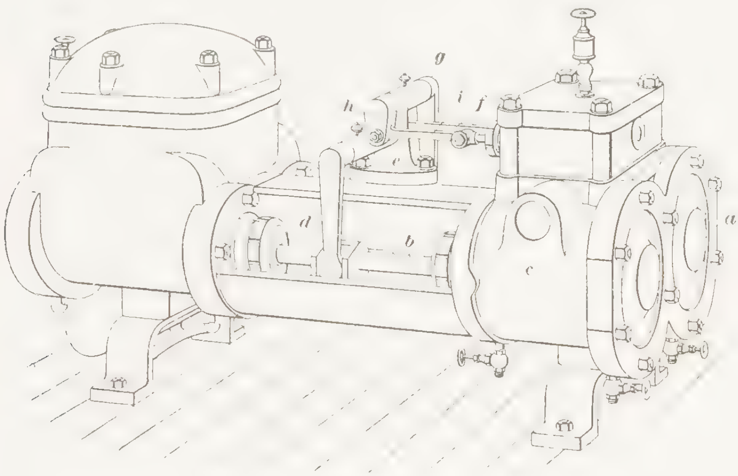


FIG. 5.

duplex pump, which shows the general arrangement of the valve motion. The two pistons always move in opposite directions, and the steam valve for the cylinder *a* is worked from the crosshead of the piston rod *b* of the cylinder *c* through the lever *d*. This lever passes through the standard *e*; it is keyed to a shaft that carries a crank in line with it at the other side of the standard, and to this crank the valve rod *f* is attached. The valve rod in turn is hinged to the valve stem. The valve of the cylinder *c* is operated from the piston rod of the cylinder *a* through the lever *g*, the crank *h*, and the valve rod *i*.

25. Fig. 6 is a sectional view through the center of the cylinder *a*, Fig. 5, and shows the construction of the steam end peculiar to the Worthington pump. Incidentally it also

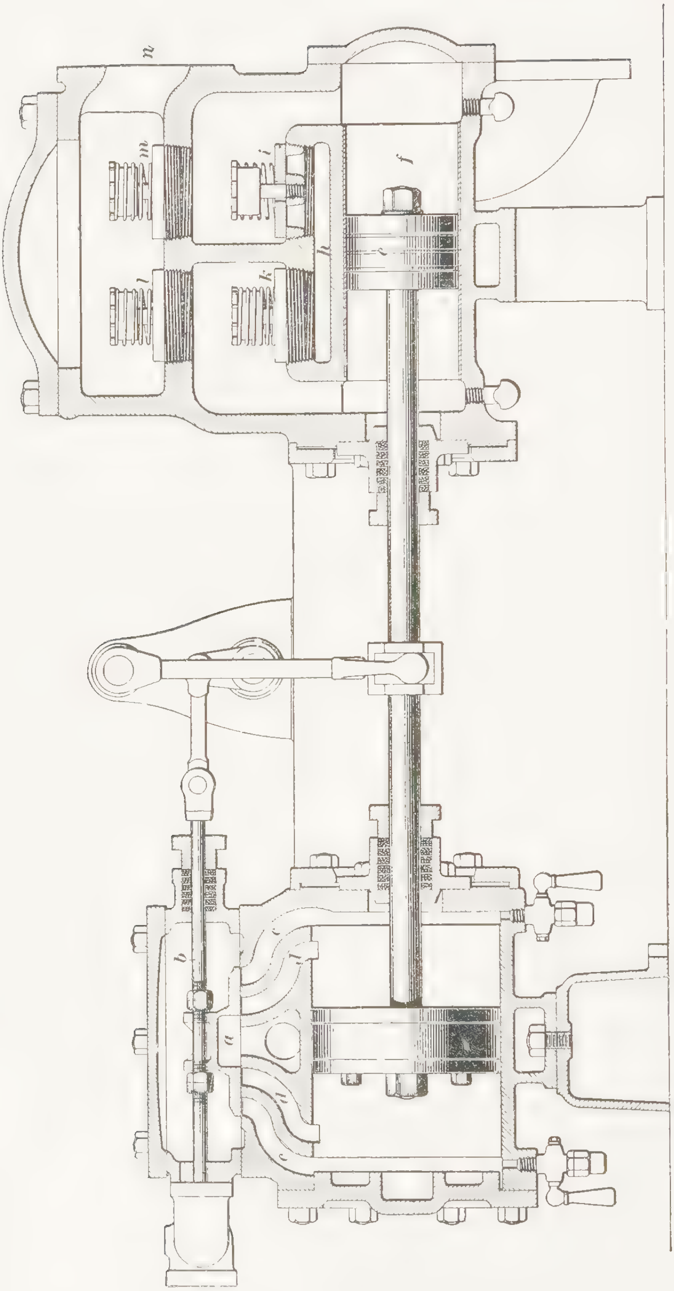


Fig. 6.

shows one form of construction of the water end of a double-acting pump. The steam valve *a* is a simple **D** slide valve operated by the valve stem *b*. There are two ports communicating with each end of the steam cylinder, of which the outer ones *c*, *c* are the steam ports and the inner ones *d*, *d* the exhaust ports. By this arrangement, when the piston approaches the end of its stroke, it covers the exhaust port and thus confines some steam in the cylinder that serves as a cushion. The valve has neither inside nor outside lap, and hence steam cannot be used expansively. The steam valve is carried along by coming in contact with check-nuts on the valve stem *b*, so placed that there is some lost motion between them and the valve. By this means the steam piston is caused to be at rest for a short time at the end of the stroke, which dwell allows the water valves to seat quietly.

26. In the water end the water is displaced by a piston *e* provided with suitable packing and working in the cylinder *f*. The water flows to the pump through the suction pipe connected to the lower nozzle and through the passage *h* to the suction valves *i* and *k*. The water is discharged through the discharge valves *l* and *m* into the discharge pipe connected at *n*. The operation is as follows: the piston *e* moving to the left, the suction valve *i* lifts and the discharge valve *m* remains closed, and water flows into the right-hand end of the cylinder. At the same time the water at the left end of the cylinder flows through the discharge valve *l*, which is lifted by the flow of water, while the suction valve *k* is kept closed. When the piston moves to the right, the suction valve *k* opens and the discharge valve *l* closes; at the same time the suction valve *i* closes and the discharge valve *m* opens. The pump is thus seen to discharge and take water during both strokes of the piston, and hence is double-acting.

27. Piston-Valve Worthington Duplex Pump.—Fig. 7 shows the steam end of a Worthington duplex pump

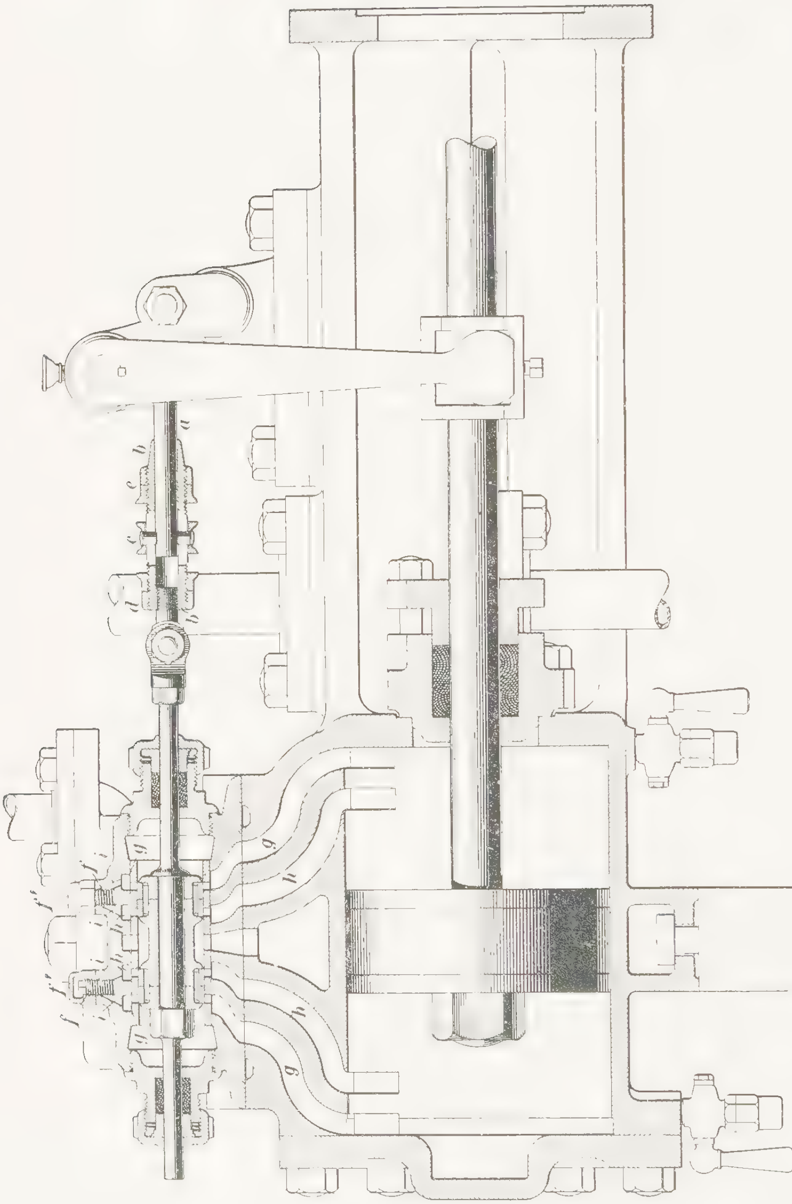


Fig. 7

in which piston valves are used instead of slide valves. The valves are operated in practically the same manner as those of the pump shown in Fig. 5, but the lost motion instead of being between the valve and stem is obtained by a special construction of the valve rod *a*. This rod is divided into two parts. The part attached to the valve stem carries a slotted yoke *b*; the part attached to the crank is free to slide within the yoke and carries a collar *c* pinned to it. The collar *c* alternately strikes against the check-nuts *d* and *e* on the yoke *b* and then carries the valve with it. The lost motion is quite large, as the valve needs to be moved but a slight amount.

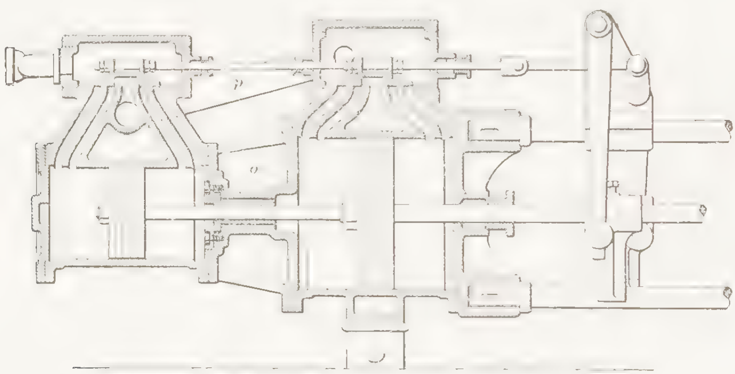
28. The length of stroke is adjusted by the use of the so-called **dash relief valves** *f*, *f*. These valves control passages *i*, *i* connecting the steam ports *g*, *g* and exhaust ports *h*, *h*, and are set by trial to the correct position and then locked with the cap nuts f', f'. The action is as follows: When the piston on its exhaust stroke covers the port *h*, no further exhaust can take place, and the steam will be compressed between the piston and the cylinder head. The location of the ports *h*, *h* is so chosen that the compression will stop the piston just short of the cylinder head at the highest speed at which the pump can operate. It is evident that when the pump is working at slow speed, the compression being the same as at high speed but the momentum of the moving parts being less, the piston will stop earlier than at high speed; i. e., the stroke is shortened. The dash relief valves prevent this shortening by providing an escape for the exhaust steam after the exhaust ports *h*, *h* are closed. It is thus seen that by them the amount of compression is regulated to suit the speed of the pump and the length of stroke is thus kept constant.

Dash relief valves are applied to pistons over 14 inches, as a general rule, and are used with slide-valve pumps as well as with piston-valve pumps. In either case they simply control a passage by which the exhaust port and steam port communicate.

MULTIPLE-EXPANSION DIRECT-ACTING STEAM PUMPS.

29. Purpose.—In the simple direct-acting steam pumps, no use can be made of the expansive force of the steam. They are, therefore, very extravagant in the use of steam, and in order to overcome this waste to a greater or less extent, many of the larger pumps are made with either compound or triple-expansion engines.

30. Compound Pump.—Fig. 8 shows a common method of arranging the cylinders for a **compound duplex** pumping



engine. The engine for each pump is made with two cylinders arranged tandem, the valves for both cylinders being driven from the same valve stem. The high-pressure cylinder is placed outside and connected to the low-pressure cylinder by a cast-iron yoke, or spacer *o*, which forms one head for each cylinder. The high-pressure piston rod passes through a sleeve in this spacer, as shown; this sleeve is held in its place by its flange being gripped between the spacer and a plate bolted on to the latter; otherwise the sleeve is free to adjust itself slightly, being a free fit in the body. The exhaust passes directly from the high-pressure cylinder through the pipe *p* to the steam chest of the low-pressure cylinder. Since there is no cut-off in either cylinder, the back pressure on the high-pressure piston is at all times equal to the pressure on the low-pressure piston, neglecting

the resistance to the flow of steam through the ports and the pipe *p*.

Since the volume of steam admitted during each stroke is equal to the volume of the high-pressure cylinder, and this steam, when exhausted, just fills the low-pressure cylinder, it is evident that the number of expansions is equal to the ratio of the volume of the low-pressure cylinder to that of the high-pressure cylinder. Also, since the length of stroke is the same for both cylinders, the number of expansions is equal to the ratio of the areas of the low- and the high-pressure piston. The usual number of expansions for small and medium sizes ranges from two to three. For large sizes four expansions are sometimes used.

31. Compound pumps are also made in which the cylinder arrangement is just the reverse from that shown in Fig. 8. In some of these compound pumps the high-pressure cylinder has no separate steam and exhaust ports; the compression and adjustment of length of stroke then takes place in the low-pressure cylinder.

32. Triple-Expansion Pump.—In triple-expansion pumping engines of the direct-acting class, the arrangement shown in Fig. 9 is sometimes adopted for the steam end. This design makes all the pistons accessible and at the same time avoids the use of a stuffingbox between the high-pressure cylinder *A* and intermediate cylinder *B*. The low-pressure piston and intermediate piston are connected by the piston rod *c*, and the low-pressure piston is connected to the high-pressure piston rod by the side rods *e*, *e* and the yoke *f*. The piston rod *c* is nicely finished and ground and works through a cast-iron bushing *g*, which is a nice fit. This bushing can move sidewise slightly so as to accommodate any want of alinement between the two cylinders. At the same time it prevents leakage of steam from the intermediate cylinder *B* to the low-pressure cylinder *C*. The low-pressure and high-pressure stuffingboxes are quite accessible. Access to the different pistons is had by removing the covers *h*, *i*, and *k*.

33. Fig. 10 is a vertical longitudinal section of the pump whose piston rod and cylinder arrangement is shown in Fig. 9 and shows the steam distribution in this form of pump. In the illustration, *A* is the high-pressure cylinder; *B* is the intermediate-pressure cylinder; *C* is the low-pressure cylinder; *d* is the high-pressure distributing valve; and *e*, *e* are the high-pressure cut-off valves. Steam enters through the center of the valve *d* and passes through the port *f* and through the cut-off port *g* into the high-pressure cylinder by way of the port *h*. The cut-off is effected by turning the rotary valves *e*, *e*. Exhaust from the high-pressure cylinder takes place through the ports *j*, *j* and thence into the high-pressure exhaust *k*, which leads to the inside of the intermediate steam valve *l*. The valve *l* is a rotary valve designed to distribute the steam exactly in the same manner as a common **D** slide valve. The intermediate- and low-pressure cylinders are not provided with cut-off valves. The exhaust steam from the intermediate cylinder passes out through the port *m* into the

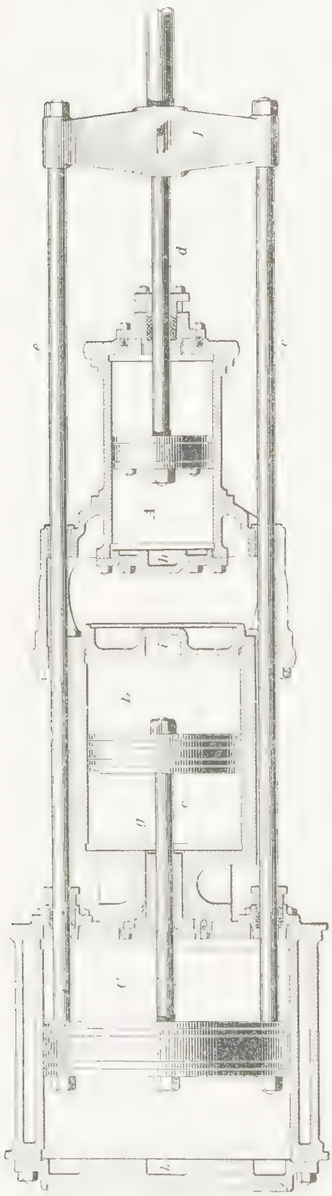
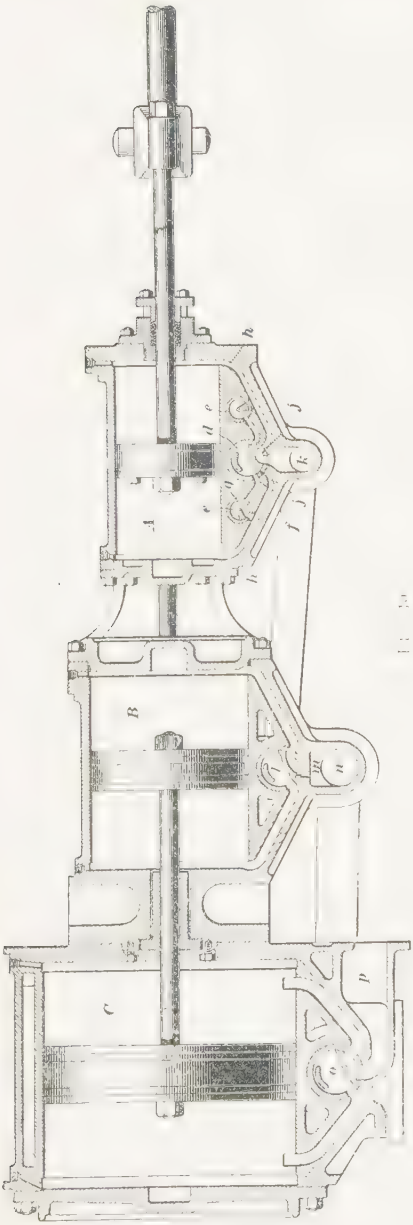


FIG. 9.



exhaust pipe *n*, and thence to the center of the low-pressure distributing valve *o*. From the low-pressure cylinder the steam is exhausted into the exhaust chest *p* and thence into the condenser or atmosphere. Dash relief valves, not shown in the illustration, are provided on the low-pressure cylinders only. The distributing valves are worked as usual from the pump on the opposite side, while the cut-off valves are worked from the pump on which they are placed.

34. Cross Exhaust. Compound duplex direct-acting pumps are occasionally provided with a so-called **cross-exhaust** connection, the purpose of which is the keeping of a more uniform pressure in the steam chests of the low-pressure cylinders than obtains otherwise. As shown in Fig. 11, it is simply a pipe *a* of ample size, which is provided with a valve *b* and connects the steam chests of the low-pressure

cylinders. The exhaust from the high-pressure cylinders *c*, *c* flows through the exhaust pipes *d*, *d* into the low-pressure steam chests *e*, *e*, but as the steam pressure there drops towards the end of the stroke, there is a diminishing of the

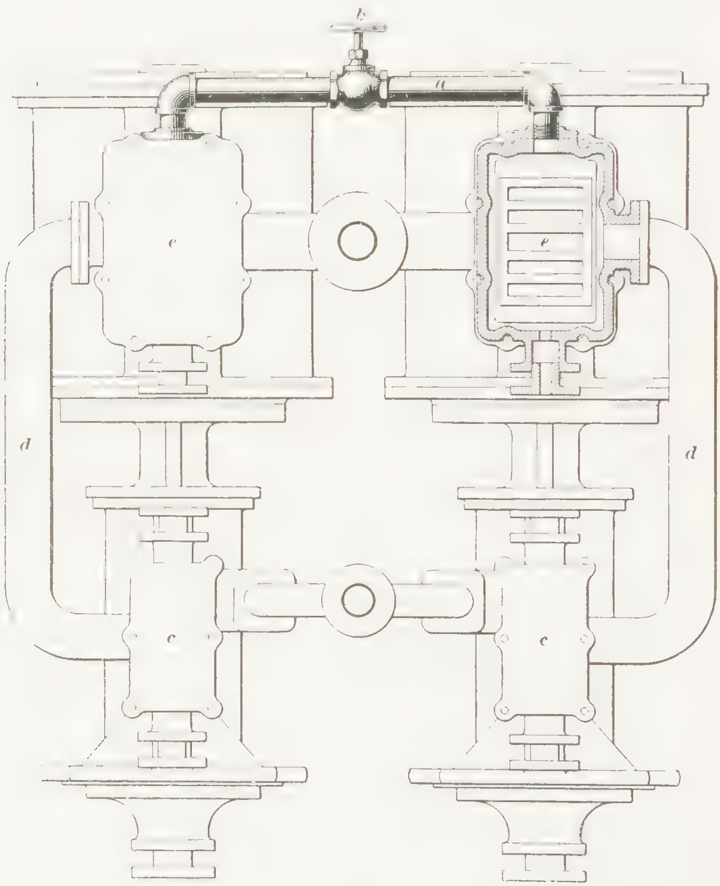


FIG. 11.

impelling force on the steam pistons of the low-pressure cylinders that tends to shorten the stroke. With the valve *b* open, the exhaust from the high-pressure cylinder of one pump can pass to the low-pressure steam chest of the other

pump just when the pressure in that steam chest commences to drop, and in consequence the pressure will be kept more uniform, which results in a steady and uniform motion.

HIGH-DUTY ATTACHMENT.

35. Purpose.—The direct-acting pump, as previously stated, is one of the simplest machines for pumping liquids, but in order to work at its best requires steam at full boiler pressure to be carried to nearly the end of the stroke. In consequence, if viewed from the standpoint of steam consumption, it is a very wasteful machine. The direct-acting pump is made more economical by making it compound or triple expansion, but even with these arrangements it is not possible to secure the high ratios of expansion which are necessary for extreme economy in the use of steam, and hence of fuel, and which are demanded in large pumping plants for commercial reasons. In the ordinary steam engine, and also in the flywheel pattern of pump, power is stored up in the flywheel at the beginning of the stroke and given out when expansion begins, in order to have a uniform turning of the engine shaft, or a nearly uniform force acting upon the water piston in case of a pump. In a direct-acting steam pump, however, there are no heavy moving parts similar to a flywheel, and hence ordinarily no uniform impelling force can act on the water piston if steam is cut off early in the stroke. This defect led to the design of the **high-duty attachment,** which is simply a device that stores up power during the first half of the stroke and gives it out again during the second half, thus allowing steam to be used expansively in the steam cylinders.

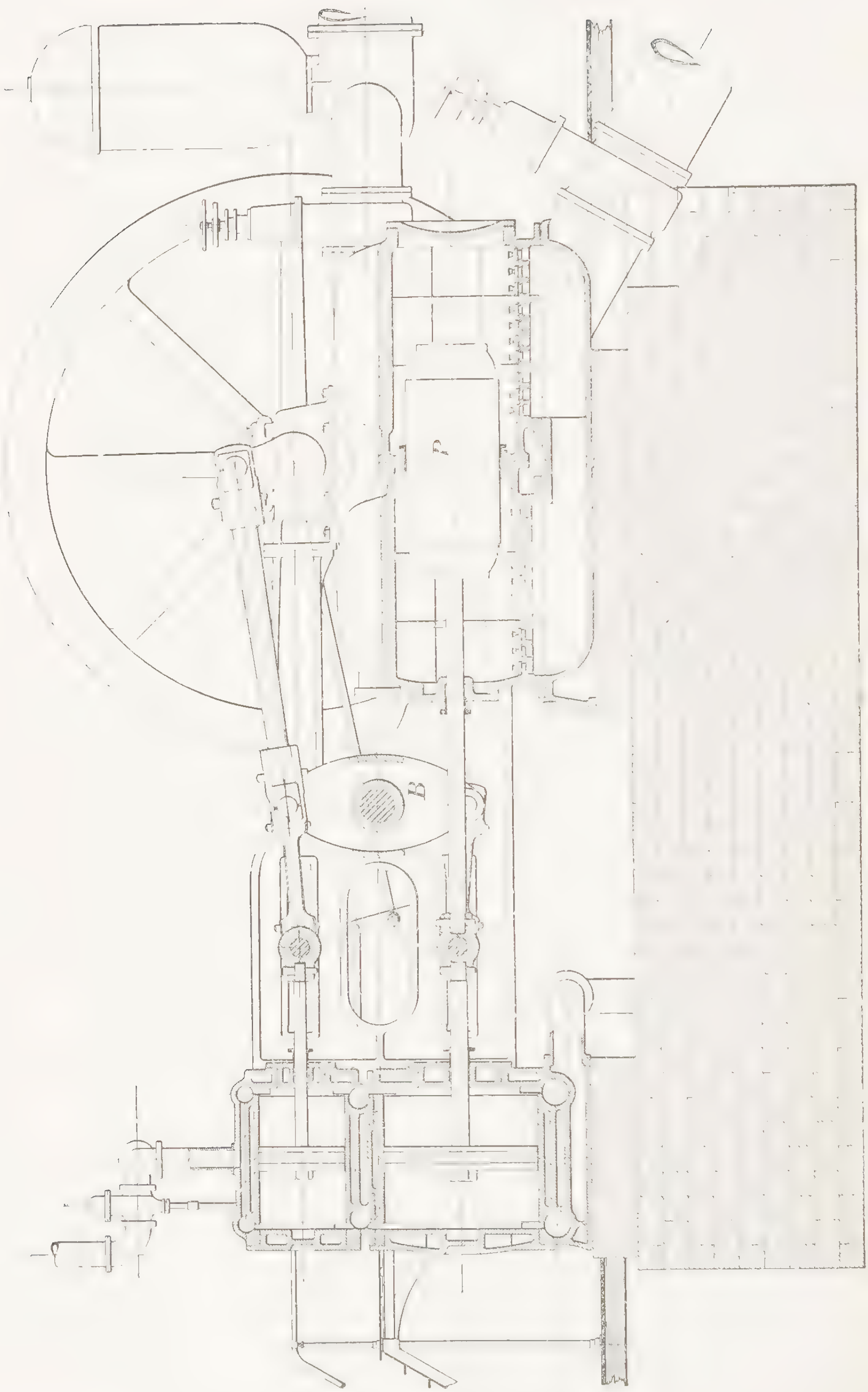
36. Construction.—The high-duty attachment in actual use was designed by Mr. J. D. Davies in 1879 and taken up and perfected by Henry R. Worthington. It is shown in Fig. 12 applied to a compound direct-acting pumping engine

fitted with Corliss valves and cutting off early in the high-pressure and low-pressure cylinders. The piston rods are arranged so as to avoid internal stuffingboxes, and, in consequence, the pistons are accessible without having to dismantle the pump. The two piston rods of the low-pressure piston and the high-pressure piston rod are attached to a common crosshead *a*, which runs in guides between the pump chambers and high-pressure cylinders. On this crosshead and opposite to each other are semicircular recesses. On the guide plates are cast two journal-boxes, one above and the other below the crosshead, equally distant from it and at the point equal to the half stroke of the crosshead. In these journal-boxes are hung two short cylinders *b*, *b* on trunnions that permit the cylinders to swing backwards and forwards in unison with the motion of the plunger crosshead. Within these swinging cylinders are plungers *c*, *c*, which pass through a stuffingbox on the end of the cylinders, and on their outer end have a rounded projection c', which fits in the semicircular recesses in the crosshead. Consequently, as the crosshead moves back and forth, it carries with it the two plungers *c*, *c*, which, in turn, tilt the cylinders backwards and forwards. These swinging cylinders are called **compensating cylinders**; they are filled with water or with whatever fluid the pump may be handling. The pressure on the plunger within the compensating cylinders is produced by connecting the compensating cylinders through their hollow trunnions with an **accumulator** *d*, the ram of which moves up and down as the plungers of the compensating cylinders move in and out. The accumulator used is of the differential type; that is, it has a small cylinder *c* filled with oil or water in which its ram moves, and above it has a much larger cylinder *d* filled with compressed air. On the top of the ram of the accumulator is an enlarged piston rod carrying a piston, which fits closely in the air cylinder. From this construction it follows that the pressure per square inch on the ram of the accumulator will be the pressure of the air in the air cylinder per square inch multiplied by the ratio between the area of the air piston and the ram

of the accumulator. The ratio of these areas is made to suit the particular service for which the pump is constructed. The pressure in the air cylinder is controlled by the pressure in the main delivery pipe of the pump, as it is connected to the air chamber *f* on the main delivery pipe.

37. Operation.—The operation of the high-duty attachment will now be explained. Suppose the pump is about to begin the forward stroke. At this time the water cylinders will be turned so as to point towards the steam cylinders, with their plungers at the extreme point of their outward stroke and at an acute angle with the line of motion of the crosshead, and with the full pressure of the accumulator load pushing them against the advance of the crosshead. As the pump plunger begins its forward stroke, each forward movement it makes changes the angle of the compensating plungers, until at mid-stroke the two plungers will stand exactly opposite each other and be at right angles with the pump plungers, in which position they can neither retard nor advance the movement of the plunger. Now, as the pump plunger passes the mid-stroke position, the compensating plungers begin to push the pump plunger along, whereas before and up to mid-stroke they resisted the movement of the pump plunger. This force increases constantly, until at the extreme end of the forward stroke, and when the compensating plungers are, as at beginning, at their most acute angle, they exert their greatest force in helping to aid the pump plunger in its outward movement. The return stroke of the pump is made under precisely the same conditions as the forward stroke. It is readily seen that at the beginning of the stroke and up to mid-stroke, work is being done in pushing the compensating plungers inward, and that after the crosshead passes the mid-position, work is being done by the compensating plungers. The effect of this is a nearly uniform force on the pump piston with a varying pressure in the steam cylinders.

38. An important feature connected with the use of the compensating cylinders is that the results obtained by their


P
B

use are independent of the speed, in which respect their action is better than that of a flywheel. The high-duty attachment in some respects also acts as a safety device, comparing its action here with that of a flywheel.

FLYWHEEL PUMPING ENGINES.

COMPARISON.

39. Although direct-acting steam pumps cannot be excelled in simplicity, low first cost, and small expense for repairs, yet they can never be extremely economical in their use of steam, even when built compound and triple expansion. While there is little doubt that a high-duty attachment will greatly increase the economy, the fact remains that at present only a limited number thus fitted are in use, and the above statement holds good for direct-acting steam pumps of the ordinary design.

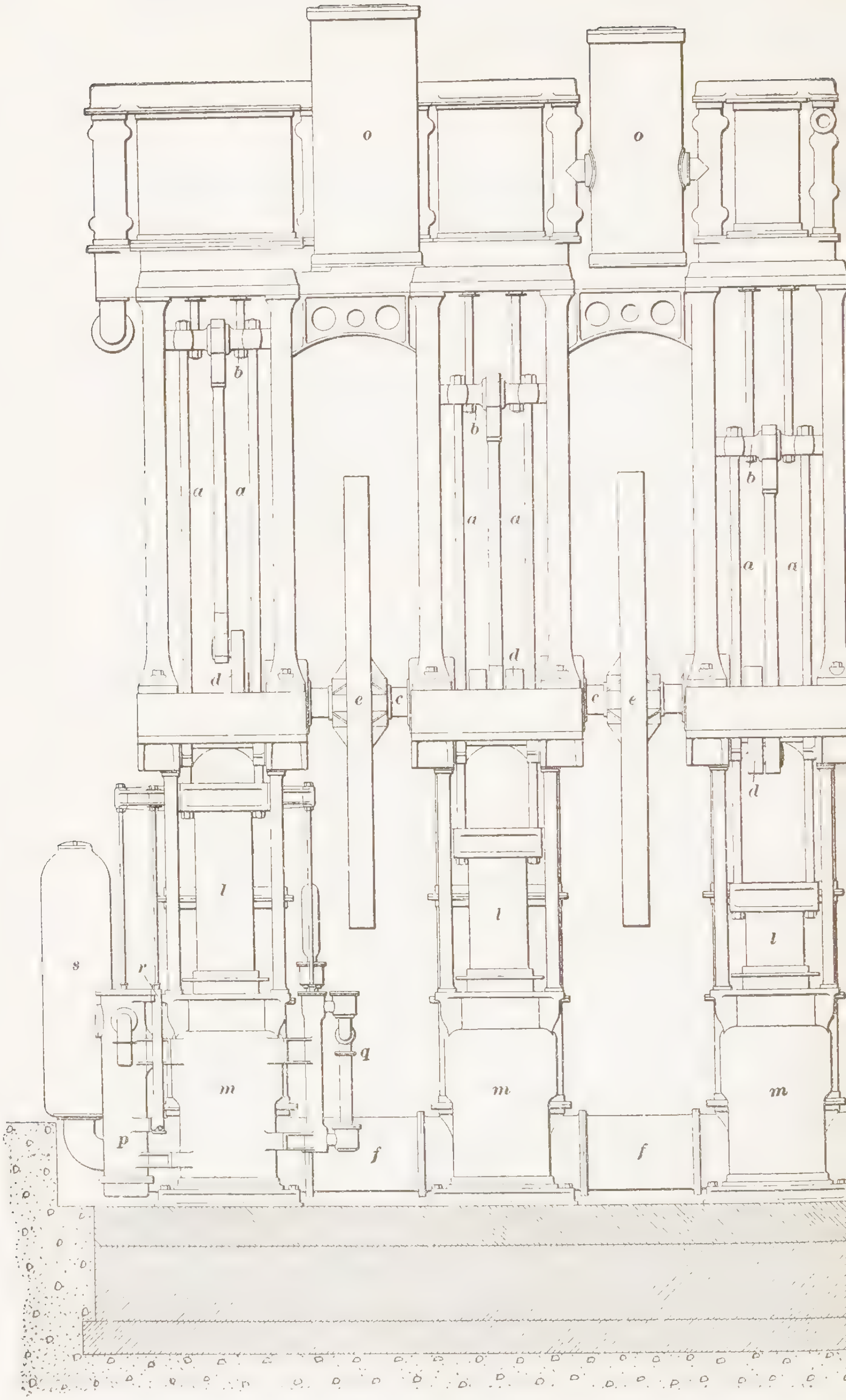
40. In large pumping stations and in many other cases where the cost of fuel is of more importance than the advantages gained from direct-acting pumps, flywheel pumping engines are often used. These are steam engines with cranks and flywheels usually designed for the particular purpose of driving the pump to which they are attached. The steam valves are driven in the ordinary way by means of eccentrics, or some approved automatic valve gear may be used to operate them. By the use of the flywheel, steam may be cut off at the most economical point in the stroke, and the surplus energy imparted to the steam piston during the first part of the stroke will be stored in the flywheel, to be given up towards the end, thus furnishing a nearly uniform driving force for the pump, piston, or plunger.

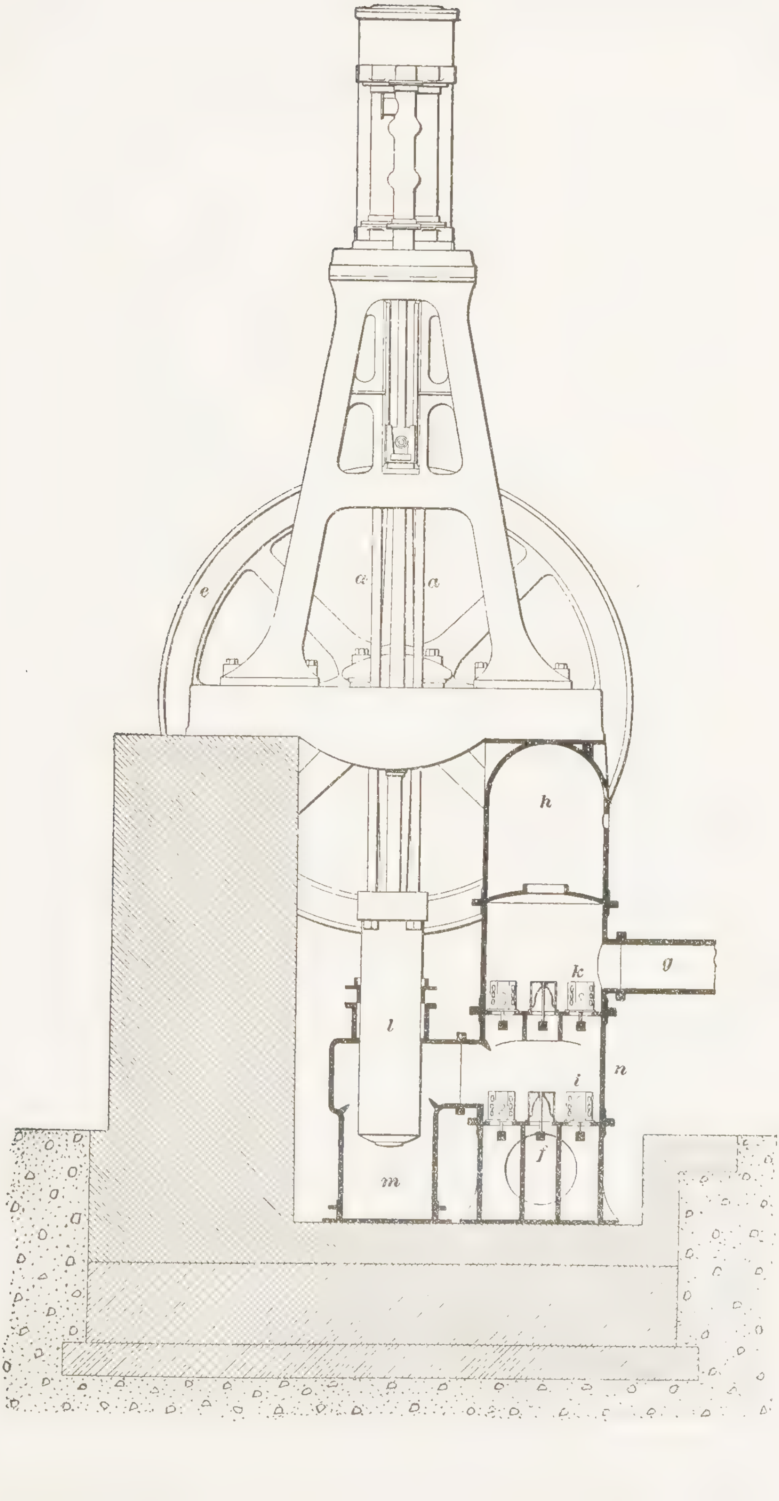
EXAMPLES OF FLYWHEEL PUMPING ENGINES.

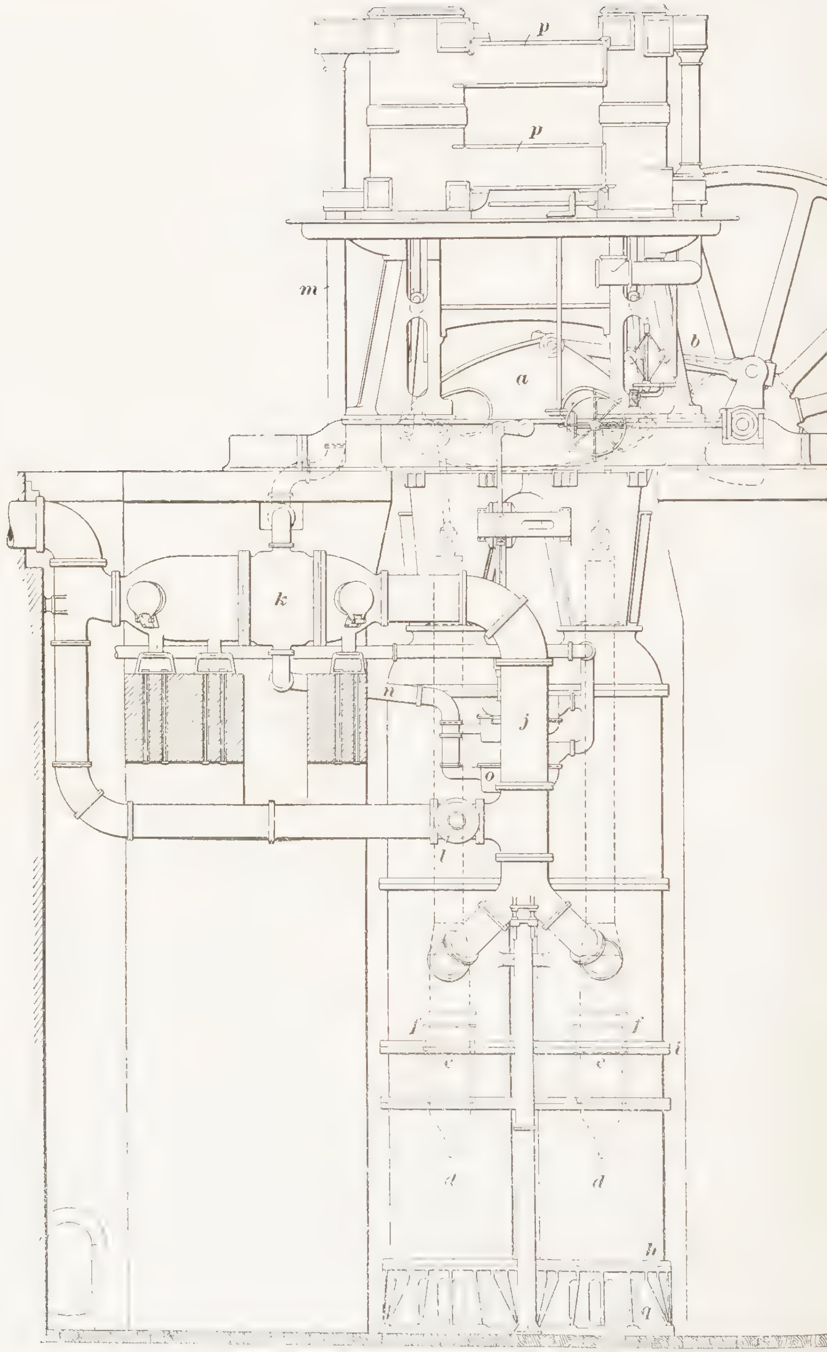
41. Fig. 13 shows a section of one side of a *Holley-Gaskill* compound pumping engine. The engine is double, the other side being like the one shown in the figure, the

two engines having a common flywheel and crank-shaft, with cranks set 90° apart. The high-pressure cylinder is placed directly over the low-pressure, with short passages between them. The connecting-rods from the two cylinders are attached to the opposite ends of a short walking beam *B*. By this arrangement the pistons move in opposite directions and the exhaust from the high-pressure cylinder passes directly to the low-pressure one. The valves are of the Corliss type, with a releasing gear for regulating the cut-off in the high-pressure cylinder. The connecting-rod that actuates the crank is attached to the upper end of the walking beam, and the rod that works the pump plunger *P* is attached to the crosshead of the low-pressure piston.

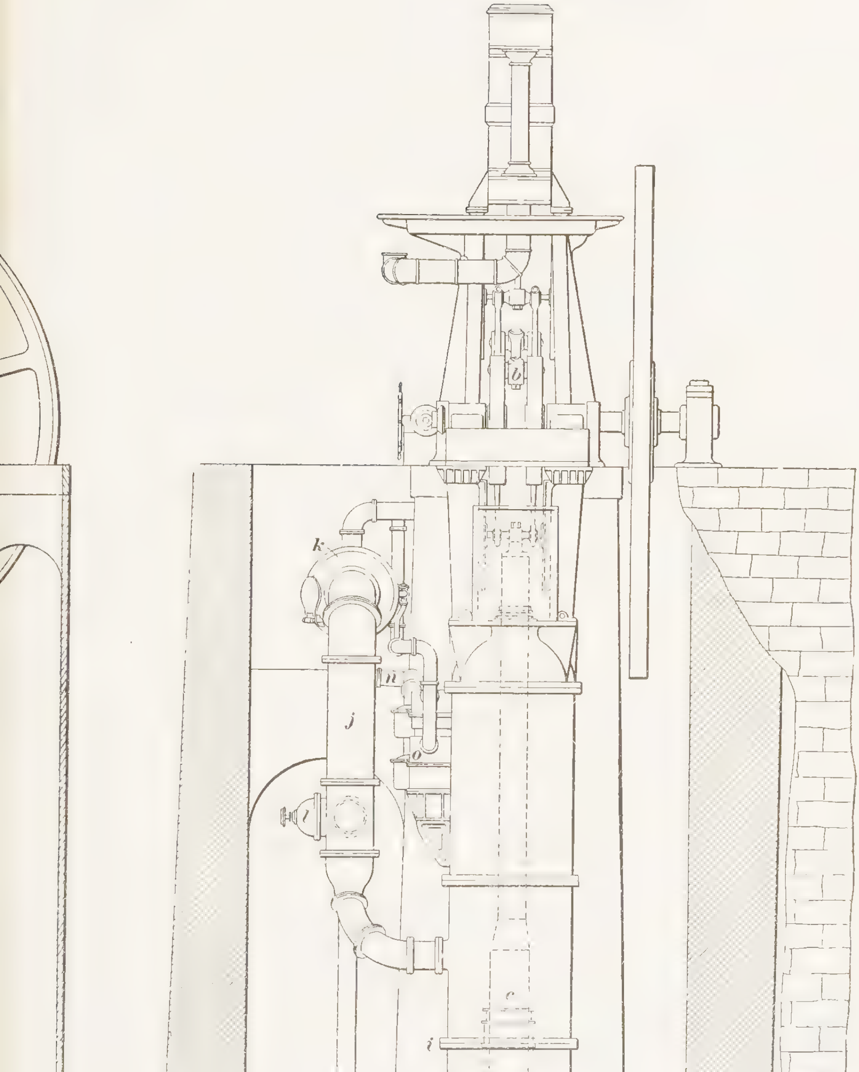
42. Fig. 14 is a front and side elevation of a modern high-duty triple-expansion pumping engine erected at the North Point pumping station, Milwaukee, Wisconsin. The engine is of the vertical inverted three-cylinder type, having the pumps in line with the cylinders, and is condensing, the condenser not being shown. Each piston is connected to a separate outside-packed single-acting plunger by means of pump rods, as *a*, *a*. There are four pump rods to each plunger, which are joined to the steam crossheads *b*, *b* and straddle the crank-shaft *c* in such a way as to allow the cranks *d*, *d* to rotate freely between them. Two flywheels *e*, *e* are employed to give uniform rotation to the machine. In the figure, *f* is the suction pipe; *g* is the delivery pipe, the delivery from each chamber being connected to a common delivery main not shown in the illustration; *h* is the air chamber; at *i* are the suction valves; at *k* are the delivery valves; *l*, *l* are the plungers and *m*, *m* the pump chambers; *n* is one of the valve chambers, the upper part of which forms the delivery air chamber *h* and also supports the fronts of the bedplates. The rear of the bedplates is supported on the masonry foundation. The steam cylinders are provided with Corliss inlet and exhaust valves on the high and intermediate cylinders and Corliss inlet valves


o
o
b
b
b
a
a
a
a
a
a
d
d
d
e
c
c
e
l
l
l
s
r
q
m
m
m
p
f
f


a
a
e
h
g
k
l
n
i
m
f



(a)



(b)

and poppet exhaust valves on the low-pressure cylinders. Large reheating receivers *o*, *o* are used between the high and intermediate cylinders and between the intermediate and low-pressure cylinders. An air pump *p* is driven directly from the plunger crossheads and serves to remove the water of condensation, etc. from the condensers. An air-charging pump *q* pumps a small quantity of air into the water in order to replenish the air supply in the air chambers. A jacket drain pump *r* drains the water from the steam jackets. A suction air chamber *s* is fitted to the extreme end of the suction pipe and prevents shocks.

43. Pumps of the design shown in Fig. 14 are used almost exclusively for high-duty municipal water-works service and are extremely economical. This type of pump has given a duty as high as 160,000,000 foot-pounds of work done per 1,000,000 British thermal units supplied to the engine.*

44. Fig. 15 shows another type of high-duty municipal pumping engine, Fig. 15 (*a*) being a side elevation and Fig. 15 (*b*) the end elevation. This pump is of the crank-and-flywheel type; the motion of the pistons is not converted into a rotary motion in the manner shown in Fig. 14, but through the intervention of a rocking beam *a*, which is rocked back and forth by the high- and low-pressure piston and is connected to the crank and flywheel by the connecting-rod *b*. This design, from its designer, is known as the **Leavitt** design. Pumps of this type have rather more parts than the type shown in Fig. 14, but they are not so high and are more accessible. The pumps are of the plunger type and are inside-packed; in the illustration, *c*, *c* are the plungers, *d*, *d* the pump chambers, and *f*, *f* the inside plunger packings. The tops of the pump chambers form delivery air chambers. The suction valves are located

* The **duty** of a pump is a measure of its performance. It will be explained in detail later.

at *h* and the delivery valves are at *i*; the delivery pipe *j* discharges the water through the surface condenser *k*, thus using the delivery water for condensation. A butterfly valve *l* controls the amount of water passing through the condenser *k*. The exhaust pipe *m* from the low-pressure cylinder enters the top of the condenser; the pipe *n* leads from the condenser to the air pump *o*. This pump is double-acting and is driven from an arm attached to the beam *a*. Two reheating receivers *p*, *p* are used to heat the steam from the high-pressure cylinder during its passage to the low-pressure cylinder. The lower ends of the pump chambers rest directly on the bottom of the pump well, which is open to the river from which the pump takes its water. The water inlets are at *q* all around the base of the pump. It will be noticed by the arrangement of the connections of the steam piston and plungers to the beam that the steam pistons have considerably more stroke than the water plungers and consequently work at a considerably higher speed, which is a decided advantage in many respects. This pump, which is located at Louisville, Kentucky, gave the remarkable duty of 151,672,000 foot-pounds of work per 1,000 pounds of dry steam used by the engine, which is the highest duty on record for any compound engine.

ROTARY PUMPS.

45. Numerous attempts have been made to replace the reciprocating motion of the piston or plunger as used in the ordinary pump by a continuous rotary motion. The results have been unsatisfactory in many cases, owing principally to the difficulty in keeping the moving parts from wearing very rapidly, thus soon producing leakage.

46. Fig. 16 shows one of the oldest and at the same time one of the best **rotary pumps.** It consists of a chamber *a* in which two toothed wheels, or disks, *b*, *b* revolve in the direction shown by the arrows. The teeth of one wheel fit

accurately into the spaces between the teeth of its mate; and, as the wheels revolve, each tooth acts as a piston that pushes a certain amount of water ahead of it, thus drawing the water from the lower part of the chamber to the upper part, as shown by the arrows. It is very important that the flat faces of these wheels, or disks, should be a good fit between the cover and the bottom of the casing or cylinder, and the edges of the teeth also a good fit against the sides of the casing. Most of the rotary pumps that have been at all successful have been modifications of the form just shown, the principal difference being in the number and shape of the teeth on the rotating disks. One of these modifications is shown in Fig. 17. In this case the disk *a* has two teeth, or wings, which act as pistons, while its mate *b* has two recesses into which the teeth on *a* fit. The shafts of the two disks are provided with outside gearing that makes their relative motion positive and always keeps them in their proper relative position.

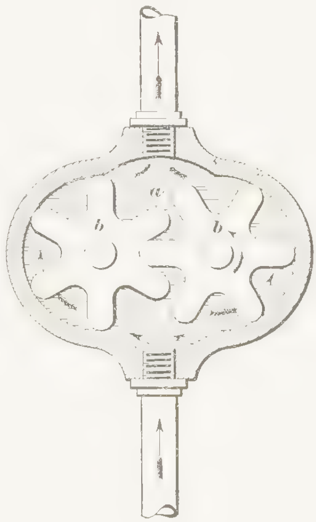


FIG. 16.

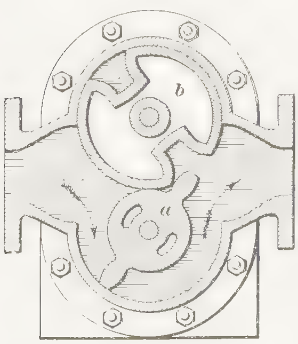


FIG. 17.

47. Fig. 18 is another modification of the rotary pump shown in Fig. 16 and gives a sectional view of **Root's cycloidal** rotary force pump. The shape of the disks or **impellers** *a*, *a* is such that the working surfaces when in contact roll upon each other. The sides of the casing are

semicircular and the impellers fit closely. The bearings in which the impeller shafts *b*, *b* run are adjustable in all

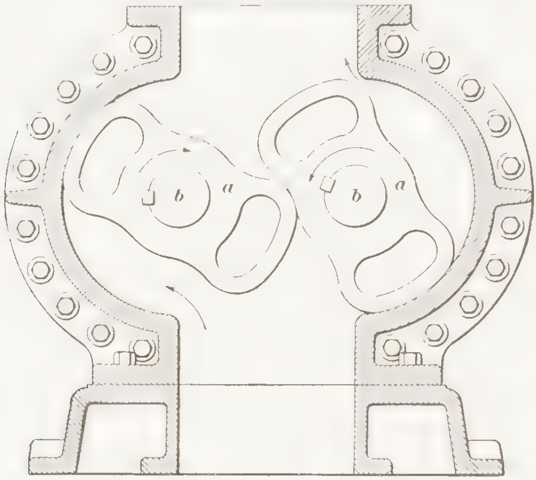


FIG. 18.

directions by means of wedges. This is claimed to be the simplest and most satisfactory rotary pump yet produced.

48. The **Quimby screw pump** shown in Fig. 19 is a rather peculiar form of a rotary pump. There are two

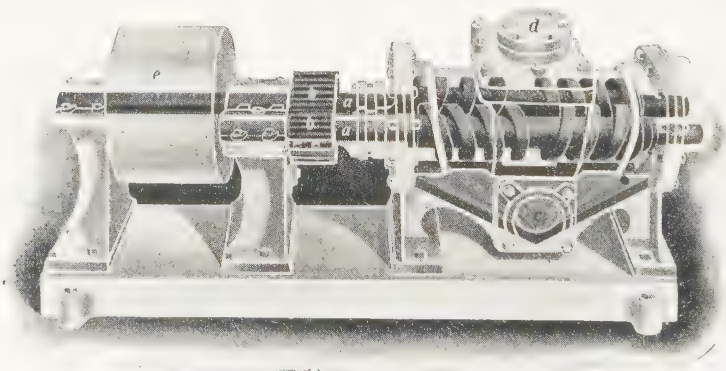


FIG. 19.

shafts *a*, *a* side by side and connected by the gears *b*, *b*. Each shaft carries a right-handed and a left-handed screw,

and the right-handed screw of one shaft meshes with the left-handed screw of the other shaft. The water coming through the suction pipe attached at *c* flows through passages in the casing to the outer ends of the screws and is drawn towards the center by the revolving screws, from whence it is discharged through *d*. The screws closely fit the pump casing and are a close running fit on each other. Since the screws are right-handed and left-handed and the course of the water is towards the center from the end of the four screws, there is no end thrust. The pump may be driven by a belt placed on the pulley *e*, or an engine or electric motor may be connected directly to it.

CENTRIFUGAL PUMPS.

49. Centrifugal pumps depend for their action on the pressure produced by the centrifugal force of a quantity of water rotated rapidly by the vanes of the pump. Fig. 20 shows two sectional views of a centrifugal pump and clearly shows its construction. The water flows through the suction inlet *a* into the chamber *b*, thus delivering the water to the inner ends of the vanes *c*, *c*, which revolve in the direction of the arrow. When the vanes are revolved, the air between them is driven out by centrifugal force, thus forming a partial vacuum. Water is forced in through the suction pipe by the pressure of the atmosphere and fills the space between the vanes. The water, of course, is made to revolve with the vanes, and the action of centrifugal force drives it outwards into the spiral-shaped passage *d*, which leads it to the discharge pipe connected to the outlet *e*

50. Centrifugal pumps are most efficient when working under low heads and are seldom used for lifts greater than 40 feet. For low heads and large quantities of water they give excellent results, and are especially useful when the water contains grit or other impurities that would destroy the pistons and packing or prevent the closing of the valves of other pumps. Since there are no valves or other

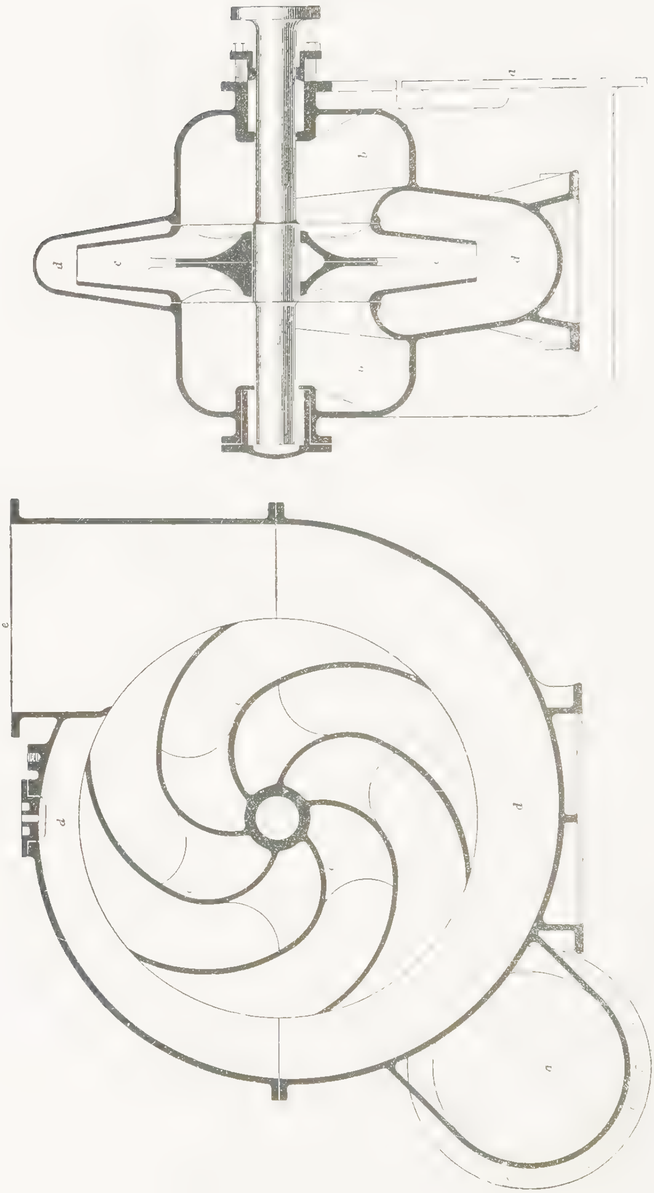


FIG. 20.

restricted passages, centrifugal pumps have been largely used in dredging machines for pumping water containing large quantities of mud, sand, and gravel; and, in fact, anything can be pumped that will be carried through the pump and pipes by a current of water. Centrifugal pumps may be belt-driven or be direct-connected to an engine or other motor.

POWER PUMPS.

51. Definition.—Pumps in which the piston or plunger is driven by a crank that receives its motion through a belt or gearing from some outside source of power are usually called **power pumps.**

52. Single Power Pumps.—A single power pump is one in which but one pump is driven by the shaft. This pump may be either *single-acting* or *double-acting.*

53. Duplex Power Pumps.—When two pumps are driven by cranks on a single shaft, the combination is called a **duplex power pump.** The discharge branches from the two pumps are generally combined in such a way that they discharge through a single pipe; and by a proper arrangement of the cranks, the flow through the discharge pipe and the power required to drive the pumps are made nearly constant. If the pumps are single-acting and the cranks are set 180° apart, the discharge from the two pumps will be the same as the discharge from one double-acting pump with the same diameter of piston and length of stroke. Duplex double-acting pumps, with cranks set 90° apart, are much used and give a very steady discharge, since, when one crank is on its dead center and its piston, consequently, is at the end of its stroke and momentarily at rest, the other piston is moving at its maximum velocity and discharging at its maximum rate.

54. Triplex Power Pumps.—Three pumps driven by cranks on a single shaft form a **triplex pump.** The most common application consists in the use of three single-acting plunger pumps with cranks set 120° apart. With

such a combination, at least one of the pumps is always discharging and one taking water from the suction pipe, and the flow is therefore continuous and nearly uniform.

55. Fig. 21 shows a type of triplex belt-driven power pump much used for feeding boilers, filling elevated tanks in

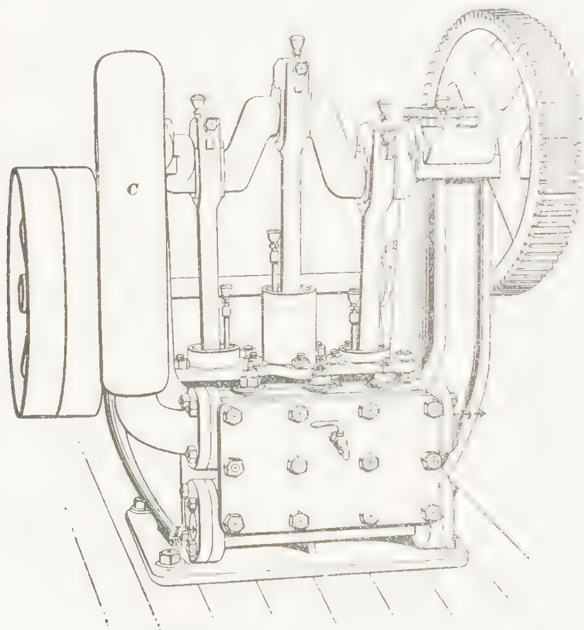


Fig. 21.

buildings, supplying hydraulic elevators, etc. It consists of three single-acting plunger pumps driven by cranks set at 120° on a single shaft. A tight and a loose pulley provide the means for starting and stopping the pump, without disturbing the engine or main shaft. The pulley shaft is geared to the crank-shaft by a pinion and spur wheel. *I* is the suction inlet, *D* the discharge opening, and *C* the air chamber.

56. Where the supply of power is steady, a belt-driven power pump is very convenient and economical for the purposes for which such pumps can be used, since they get their

power with the same degree of economy as the engine by which they are driven; they are also simple in construction and easily operated.

57. In locations where there is no steam or other power directly available, or where the use of the pump is so intermittent that a steam plant will not be economical, or where the cost of supplying steam is too great, power pumps driven by electric motors may be used to advantage. Small pumps driven by windmills, hot-air engines, gas engines, etc. are much used for supplying water to buildings that have no connection with public water works. Small, single-acting plunger pumps are most commonly used with these methods of driving, although double-acting pumps are sometimes used. Where water-power is available, pumps for city water works or for supplying manufacturing establishments are often driven by waterwheels.

MINE PUMPS.

SERVICE.

58. Pumps intended for the drainage of mines are probably subjected to the hardest usage of any. The water to be pumped is generally gritty and frequently it contains a large percentage of acids; a very high pressure must generally be pumped against and the pump has to run almost continuously for long periods at the full limit of its capacity. In most cases the mine is located quite remote from supplies; the pump of necessity is underground and in a rather limited space; it is generally of vital importance that the pump be kept running in order to prevent the drowning out of the mine, and for the same reason it is desirable that all wearing parts be very accessible so that repairs can be made in the shortest time. Furthermore, it is desirable that the pump continue at work even when covered entirely with

water. The exigencies of the service have led to designs of pumps especially suited for the work. While they do not differ essentially from ordinary pumps, they have generally a different arrangement of water end. Nearly all mine pumps are of the plunger pattern, the plunger pump, by reason of the ease with which leakage can be stopped, being best adapted for high pressures.

59. Mine pumps are either pit pumps, direct-acting steam pumps, or power pumps. By a pit pump is meant a pump having its water end located at the bottom of the mine and connected to a steam engine or other motor at the surface by rods. Pit pumps are the oldest type of mine pump and are still used to some extent.

TYPES OF MINE PUMPS.

CORNISH PUMPING ENGINE.

60. Until within comparatively recent times, the so-called **Cornish pumping engines** have been the only ones used for removing the water from the mines. This engine was invented by Watt for use in the mines of Cornwall and was the first really effective steam engine made. An illustration of a Cornish pumping engine is shown in Fig. 22. The cylinder *A* is single-acting; that is, the steam acts only on one side of the piston. The piston rod *B* is connected to the walking beam *C* by a link *R*. In Cornish pumping engines, the steam is admitted through the valve in *I* to the top of the piston and forces it down towards the bottom of the cylinder. The weight of the pump rods and other moving parts in the shaft, which parts are called the **pit work,** is sufficient to raise the piston to the top of the cylinder when the steam on the upper side of the piston is put in communication with the lower side. The cylinder *A* is steam-jacketed; that is, the cylinder walls are hollow and

filled with steam in a manner similar to the water-jacket of an air compressor, the steam entering through the pipe *K*.

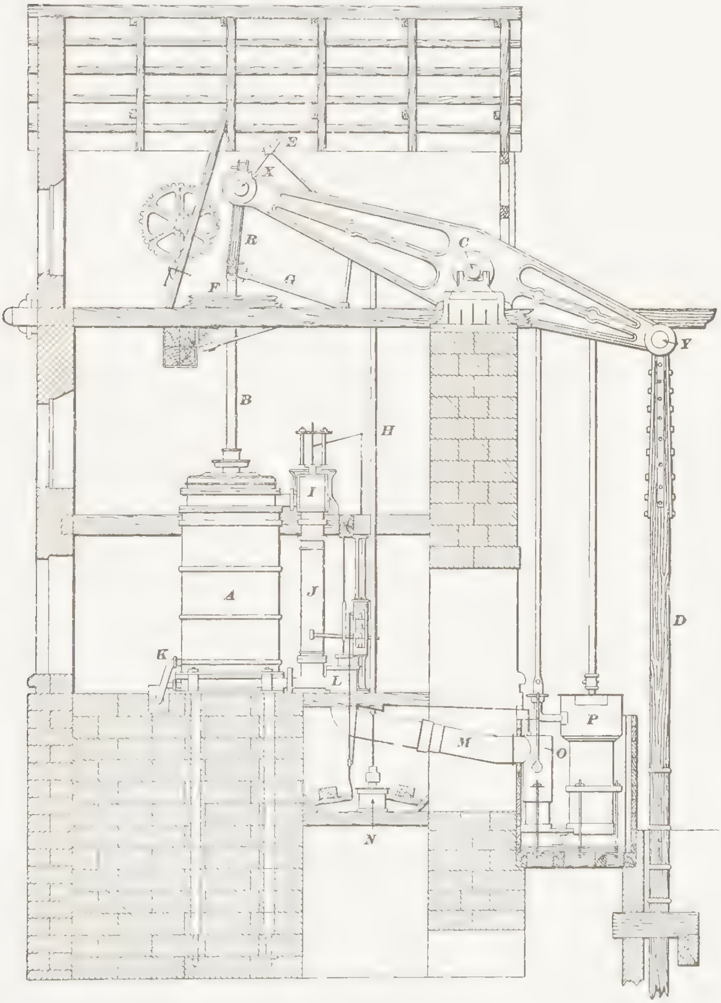


FIG. 22.

61. The action of the pump is as follows: Steam is admitted to the upper side of the piston through a valve in *I*,

which is operated by means of a tappet rod *H*. The steam is of high pressure and forces the piston rod downwards and at the same time raises the pit work. This gathers momemtum while coming upwards, and the steam is cut off, expanding during the rest of the stroke. Just before the end of the stroke, what is termed an *equilibrium valve*, also located in the casing at *I*, opens and allows the steam in the upper end of the cylinder to communicate with that in the lower end. The two pressures being thus balanced, the heavy pit work causes the right end of the walking beam *C* to descend, raising the piston to the top of the cylinder again. The exhaust valve is located at *L*. When this is raised, the exhaust steam flows through the pipe *M* into the condenser *O*. *P* is a small pump used in operating the condenser. *E* is a catch intended to act in case the valves should fail to work. The piston rod passes between two blocks, of which *F* is one, the other being opposite. If the left end of the walking beam should descend too far, a crosspiece on the catch rod *E* is caught by the blocks *F* and prevents any further downward movement of the piston.

BULL ENGINE.

62. In Fig. 23 are shown two views of a Cornish Bull engine and pump. This style of pumping engine is made by many firms and differs but very little in regard to details. Here the walking beam is dispensed with and the cylinder is placed directly over the shaft, the pit work being attached to the piston itself. In this case also the cylinder is single-acting, the steam being admitted below the piston instead of above it, as in the engine described in Fig. 22. The condenser is usually omitted in this class of pumps, the steam exhausting directly into the atmosphere. In case the weight of the pit work should be greater than necessary to force the water up the required height, the extra weight is counterbalanced.

63. The Bull pumping engine possesses several advantages over the Cornish pump. The heavy walking beam

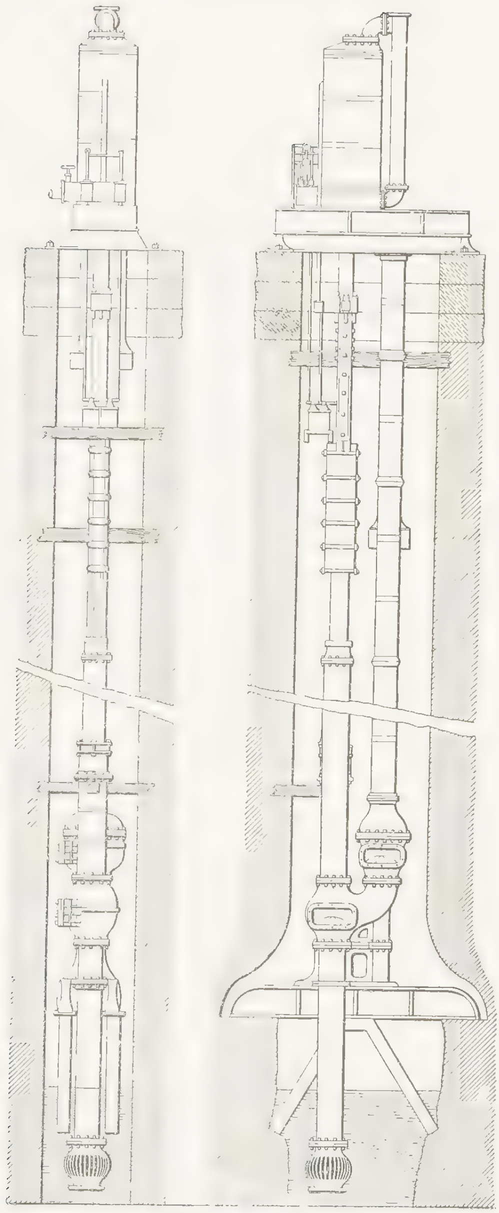


FIG. 23.

and its connections are dispensed with; this lessens the first cost; the friction is greatly reduced; the advantage of having a direct-acting engine is also obtained. The principal disadvantage is that the pump being directly over the shaft, takes up a great deal more room where space is necessary than the Cornish pump.

64. Cornish and Bull pumps both use steam expansively. They do not have flywheels to absorb the energy of the early part of the stroke and give it out again at the end, but utilize the heavy pit work to accomplish the same purpose. The number of expansions ranges from four to ten; that is, the steam is cut off from $\frac{1}{4}$ to $\frac{1}{10}$ stroke. When using more than six expansions ($\frac{1}{6}$ cut-off), the strain produced on the machinery becomes very heavy, and the resulting wear and tear of the machinery more than makes up for the increased economy in the use of steam. Many engineers claim that four expansions are the most economical.

PIT-PUMP ARRANGEMENT.

65. The arrangement of the pump driving mechanism, pit work, and balancing mechanism in a deep mine shaft is shown in Fig. 24. In this case an ordinary horizontal engine is used at the surface, which works the pumps through the intervention of a gear train and a connecting-rod and crank, the connecting-rod being attached to the bell-crank *A*. On account of the great length of the rod (over 1,600 feet), its weight added to the weight of the plungers is considerably more than the weight of the water column; hence, to save the extra power which would be required to be used in raising this extra weight, it is counterbalanced. A counterbalance weight *X* is placed on one end of the bell-crank *A*; two other bell-cranks, *B* and *C*, are located down the shaft, one end carrying the counterweight *X* and the other end being connected to the pump rod by means of a link and the cast-iron offsets *D* and *F*. The water is raised by four lifts, the first, to *K*, being 360.8 feet, and the other three 328 feet each. In this particular instance, the water is

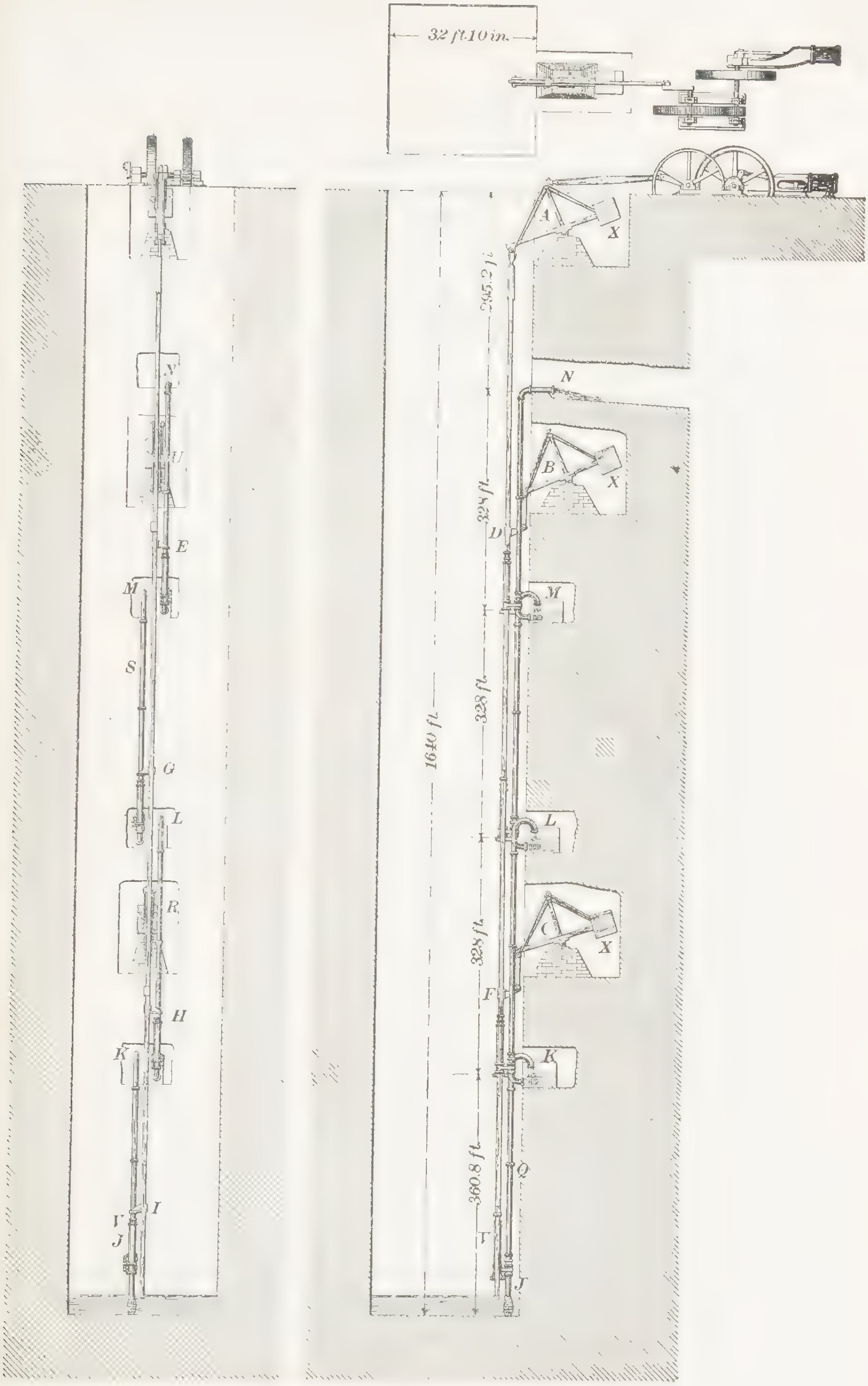

32 ft.10 in.
A
X
295.2 ft.
N
U
B
X
328 ft.
D
E
M
M
S
328 ft.
1640 ft.
G
L
L
R
C
X
328 ft.
F
H
K
K
Q
360.8 ft.
I
V
V
J
J

FIG. 24.

discharged into a tunnel *N*, about 300 feet below the surface. The pump rod goes straight down the shaft and the discharge pipes are placed alternately on each side of it. *J* is a suction pipe. *I* is a bracket, one end of which is attached to the pump rod and the other end to the pump plunger *V*. On the down stroke, the water is forced out of the pump cylinders and up the pipes *Q*, *R*, *S*, and *U*, discharging at *K*, *L*, *M*, and *N*. The same pit work and pump arrangement may be and is used for Cornish and Bull pumps.

66. The use of a geared engine possesses several advantages over the Cornish or Bull pumping engines. The flywheel permits a more even distribution of the power. The length of the stroke is always the same, and there is no danger of damage caused by the piston being blown through the cylinder head, should the valve gear refuse to work.

WATER END OF PIT PUMPS.

67. Comparison of Lifting and Force Pumps.—The water end of a pit pump may be a lifting pump or a force pump. The lifting pump is generally considered inferior to the force pump (which latter is almost invariably of the plunger pattern) for mine work.

It is easier to specify the objections to lift pumps than to state their advantages over the plunger pumps. The pump rod, being necessarily inside of the delivery pipe, reduces the effective area of pipe and increases the friction of the water to some extent, owing to the added surface rubbed against. The rods are concealed and cannot be inspected without removing the entire rod. Not only do the bolts and rods sometimes break, thus rendering their recovery difficult, but the bolts will wear against the stocks, causing loss of power by friction and destroying the pipes. Lift pumps are not so liable to sudden injurious strains as the plunger pumps.

The plunger type of pumps is superior to the lift pump in nearly every respect for very high lifts with the accompanying heavy pressure or when dirty water is being raised.

When pumping against a heavy pressure, it is impossible to keep the piston of lift pumps tight and prevent the water from leaking. The piston and cylinder of the lift pump must in every case be a perfect fit and be truly cylindrical. With plunger pumps, on the contrary, the rod passes through a stuffingbox, and the plunger may or may not fit the cylinder. When pumping dirty water, the grit comes in contact with the surface that the piston of a lift pump is constantly traveling over and destroys both the cylinder and piston very rapidly; whereas, the plunger has to be kept tight at only one permanent place, and the dirt cannot very well get at the surface of the packing on which the plunger or plunger rod rubs. Every part of a plunger pump can be readily examined and repaired without being obliged to take down the whole apparatus.

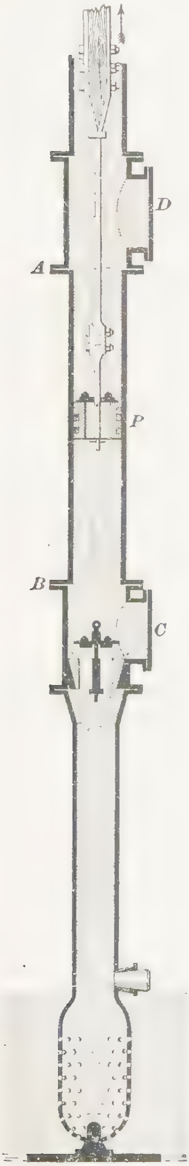


FIG. 25.

68. Example of a Lifting Pump. In Fig. 25 is shown a section of a lifting pump for use in mines. The pump consists of a series of pipes connected together, of which the lower end only is shown in the figure. That part of the pipe included between the letters *A* and *B* forms the pump cylinder in which the piston *P* works. The part above the highest point of the piston travel is the delivery pipe, and the part below the lowest point of the piston travel is the suction pipe. When speaking of these parts as applied to mine pumps, the delivery pipe is usually termed the **working barrel**, and the suction pipe the **wind bore.**

In mine pumps, the lower end of the wind bore is pear-shaped and perforated

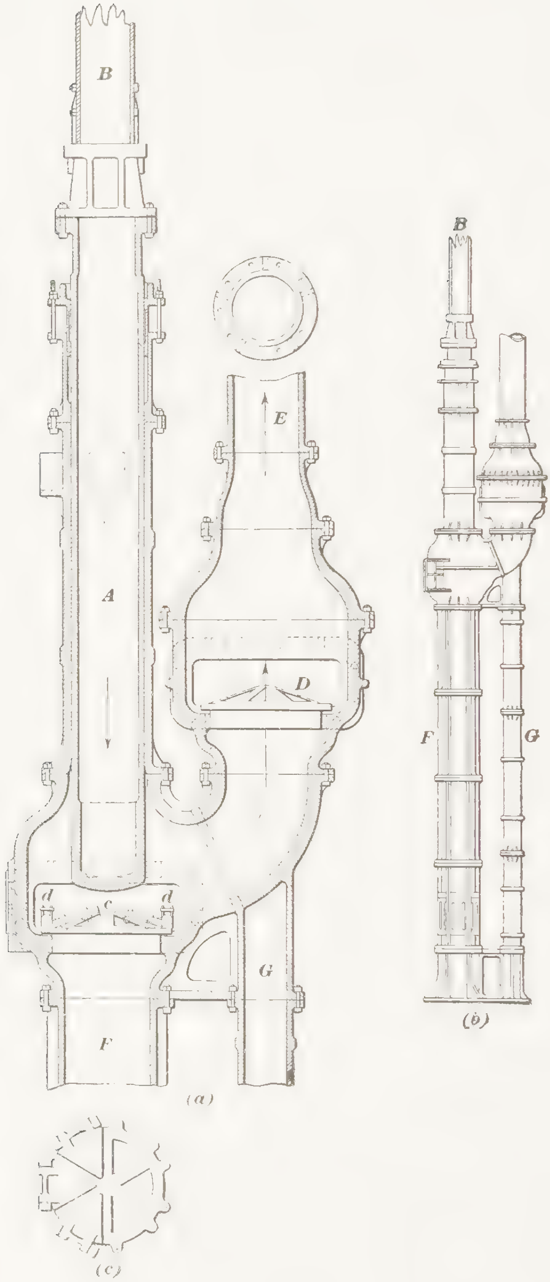


FIG. 26.

with many small holes to keep solid matter in the water from entering the pump and destroying the valves. In some cases, the pear-shaped end is covered with gauze for the same purpose. A bonnet *C* may be removed to allow the suction valve to be repaired, and a bonnet *D* gives access to the piston and its valves The pump rod is made of wood strapped with iron and is connected to the piston in the manner shown by the illustration.

69. Example of a Force Pump.—Fig. 26 shows one design of a force pump of the plunger type as used for a pit pump, Fig 26 (*a*) being a section showing the pump cylinder and valves, and Fig. 26 (*b*) showing an elevation of the whole water end drawn to a smaller scale. The plunger *A* is hollow, the weight of the heavy rod *B* and connections being sufficient to raise the water to the required height.

Suppose the plunger to be on the down stroke; the valve *c* is then closed and the water filling the pump cylinder is forced through the valve *D*, which it opens, and up the delivery pipe *E*. When the plunger reaches the end of its stroke and begins its return, the weight of the water forces the valve *D* to its seat, retaining the water above it in the discharge pipe *E*. As the plunger moves upwards it leaves a partial vacuum behind it, causing the water to rush up the suction pipe *F*, lift the valve *c*, and fill the pump cylinder. The plunger makes another downward stroke and the above process is repeated. A support *G* is attached to the delivery pipe, the lower end resting on a foundation. This is necessary, since the great weight of the water in the discharge pipe and the weight of the pipe itself would break it off at the bend unless supported in some such manner; otherwise, the thickness of the metal around the bend would necessarily be enormous.

70. A top view of the valves is shown in Fig. 26 (*c*). They consist of six triangular valves arranged in a circle, with their apexes pointing towards the center. These six valves turn upwards on hinges and are prevented from going too far by the projection *d*; see Fig. 26 (*a*). Three

of the valves have been removed so as to show the amount of valve opening that they give. When the valves are open, they form an angle of about 45° with their position when closed.

SINKING PUMPS.

71. Purpose.—When putting down a new shaft or deepening an old one, the so-called **sinking pump** is used to drain the water from the shaft bottom so that the work may proceed. These pumps must necessarily be portable and are suspended by a chain attached to eyebolts in the pump. They are also provided with wrought-iron clamps, by means of which they may be attached to the timbers in the shaft when it is desired to fix them in position temporarily. Hence, as the shaft gets deeper, the chain may be lengthened out, an extra joint placed on the upper end of the delivery pipe, and it is again ready for business. The sinking pump is subjected to the hardest usage of any mine pump. The water pumped is invariably gritty and often acid. The water trickling down on the pump from above carries mud along with it and so completely covers the pump that it is hardly distinguishable at times from the soil itself. Notwithstanding all this, a sinking pump must work night and day, often up to the limit of its capacity, and its failure, even for a day, at a critical period may flood a shaft which would require a week or more to recover.

72. Steam Sinking Pump.—In Fig. 27 a Cameron sinking pump is illustrated. This pump meets all the conditions required of a sinking pump and is a favorite with mine operators. There is no outside valve mechanism whatever, and nothing short of actual breakage of the pump itself or of the steam, suction, or delivery pipe can prevent the pump from working. The manner of suspending it from a chain is shown in the illustration, also the method of attaching it to the shaft timbers. In order to more clearly show the working of the valves and plunger, a partial section

of the pump is given in the figure. The pump has one plunger, but is double-acting by reason of its peculiar construction. It will be noticed that leakage past the plunger *A*

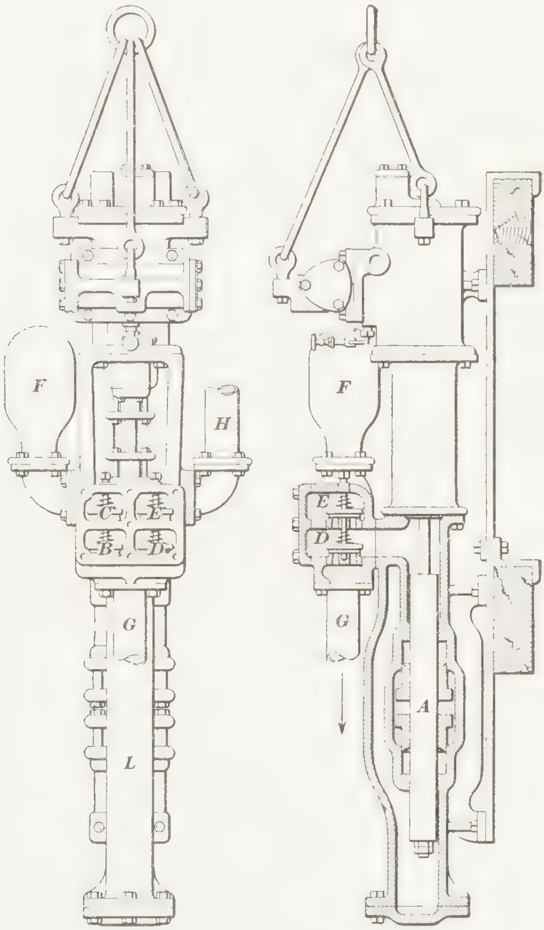


Fig. 27.

is prevented by two stuffingboxes and glands placed in the center of the pump cylinder; a pump having this arrangement is said to be **center-packed.**

The action of this pump is as follows: Suppose the plunger to be moving downwards. The water is forced out of the chamber *L*, which communicates with the delivery pipe *H* by means of the valve *C*, and lifts *C*, thus flowing into *H*. As the plunger moves down it leaves a vacuum behind it; the water in the shaft rushes up the suction pipe *G*, raises the valve *D*, and fills the upper part of the plunger cylinder. When the stroke is reversed, the valves *C* and *D* close, and the valves *E* and *B* open, the water being forced up the pipe *H* through the valve *E*, and the chamber *L* is filled through the opening of the valve *B*. *F* is the air chamber. The section shown by the view on the right is taken in a rather peculiar manner, the greater part being taken through the center line of the engine so as to show the plunger, stuffingboxes, etc., and the part showing the valves being taken on the center line of the valves *E* and *D* of the view on the left.

73. Electric Sinking Pump.—While most sinking pumps are steam-operated, electrically driven sinking pumps are also used. Fig. 28 shows a duplex electric sinking pump of the center-packed type, the stuffingboxes being shown at *H*. The two plunger rods *E* and *F* operate the plungers. A clamping piece *D* is used for attaching the pump to the shaft timbers; an eyebolt *G* is used for suspending the pump from a chain. The water enters through the suction pipe *B* and leaves through the discharge pipe *A*.

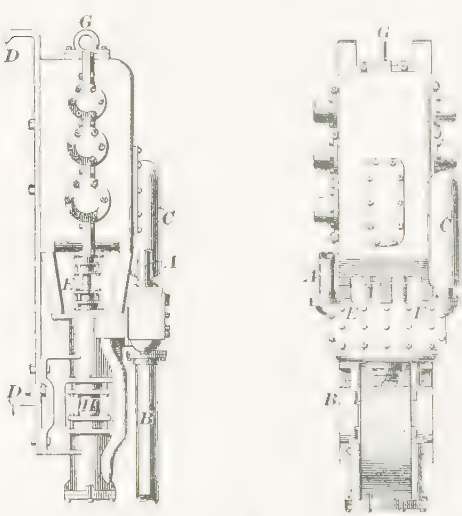


Fig. 28.

An air chamber C is fitted to the valve chamber. The electric motor is within the water-tight casing above the water end and is protected by it, so that the pump can work just as well under water as above it.

DIRECT-ACTING STEAM PUMPS FOR MINE WORK.

74. Pumps Used.—Direct-acting steam pumps used for mine drainage are almost invariably of the plunger pattern. Most of them are duplex, but a number of single double-acting steam pumps are in use. Formerly, all the mine steam pumps were simple direct-acting pumps, but of late years compound and even triple-expansion pumps have grown in favor, and even crank-and-flywheel pumps driven by compound Corliss engines are now extensively used on account of their superior economy. Most of the pumps are of the *double-plunger type*, there being two plungers to each water cylinder, and the stuffingboxes are located on the outside, thus making the pumps *outside-packed*. Some mine pumps are *center-packed* and use but one plunger for each water cylinder.

75. Simple Double-Plunger Pump.—Fig. 29 shows a side view of a simple direct-acting single mine pump of

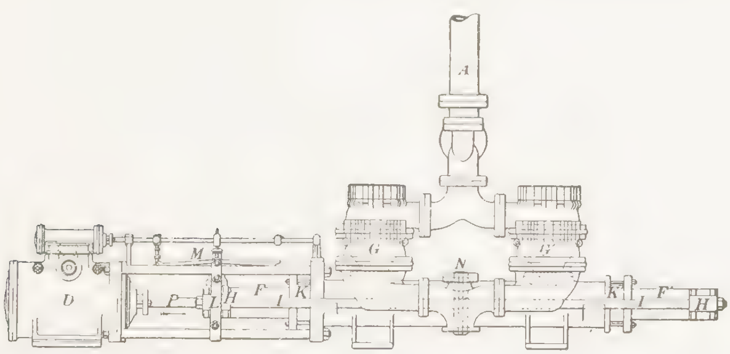
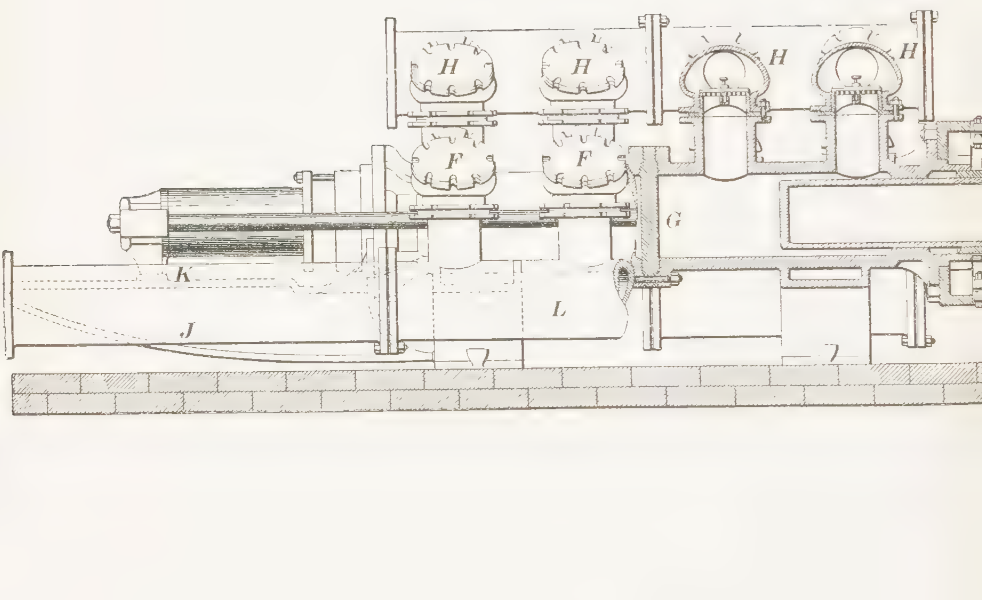


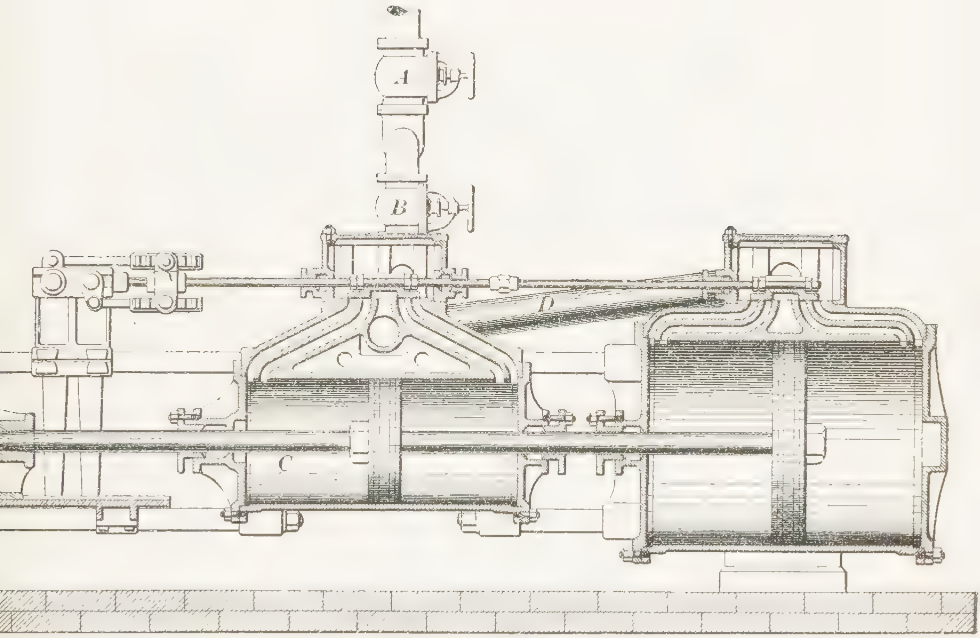
FIG. 29.

the double-plunger type. The two plungers F and F' carry yokes H, H at their outer ends and are tied together by

side rods, as *I*. The plunger *F* is attached directly to the piston rod *P*. Suppose the steam piston in *D* to be moving to the right; the plunger *F* is then forcing water into the chamber *G* and up the discharge pipe *A*. Since the plunger *F'* is moving out of the water cylinder (it will be understood that the cylinders in which *F* and *F'* work are divided by a water-tight partition at *N*), water flows in through the suction pipe, and when the pump makes its return stroke, *F'* does the forcing while water flows into the cylinder in which *F* works. It is thus seen that by the use of two plungers connected as shown, the pump is made double-acting. Stuffingboxes *K* and *K'* are used for packing the plungers.

76. Compound Double-Plunger Pump.—The internal arrangement of a double-plunger pump is shown clearly in Fig. 30, which is a sectional view of one side of a Jeanesville compound duplex mine pump designed for heavy pressures. The section shows one of the plungers *E* working inside its chamber; *G* is the partition that separates the two chambers. The outer end of each plunger is supported by a shoe *K* working on a slide *J*. *L* is the suction pipe with branches leading to the suction valve chambers *F*, *F*; the discharge valve chambers *H*, *H* connect with the discharge pipe shown just over the pump cylinders. As shown in the figure, there are two suction and two discharge valves for each plunger. The usual arrangement for pumps of this size is to have a great number of small valves instead of a few large valves, as shown, but for mine work the sulphur in the water destroys the valves rapidly and the large valves are more quickly and cheaply replaced. *A* is the main and *B* the auxiliary throttle; *C* is the high-pressure cylinder, from which steam goes to the low-pressure cylinder through the pipe *D*. The valve gear of the steam end is practically the same as that of Worthington pumps, and the steam valves of one side are operated from the piston rod of the other side. Steam is carried full stroke in all cylinders.


H
H
H
H
F
F
G
K
L
J


A
B
C
D

77. Triple Expansion Center-Packed Pump.—Fig. 31 is a side view of one side of a triple-expansion duplex Worthington mine pump having plungers which are center-packed.

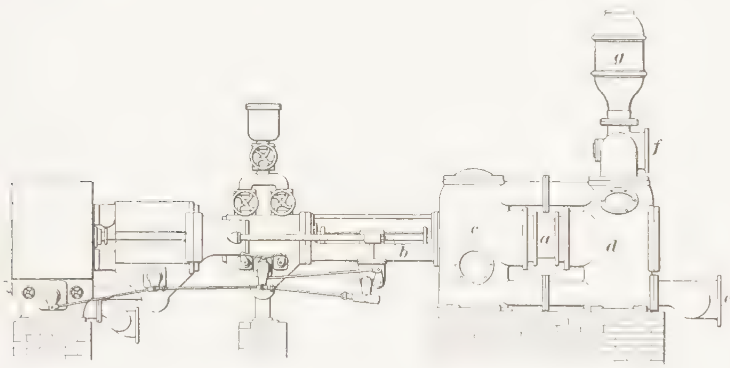


FIG. 31.

The type of water end used with this pump has been given the name of **Scranton type** by the makers. Sectional views of the steam cylinders of this pump have already been given in Figs. 9 and 10. The plunger *a* is connected to the piston rod *b* and works in the pump chambers *c* and *d*, which have the suction valves on the bottom and the delivery valves on top. The suction pipe is connected at *e* and the delivery valve at *f*. An air chamber *g* on the delivery absorbs shocks and promotes a steady delivery. The pump is double-acting.

FLYWHEEL PUMP.

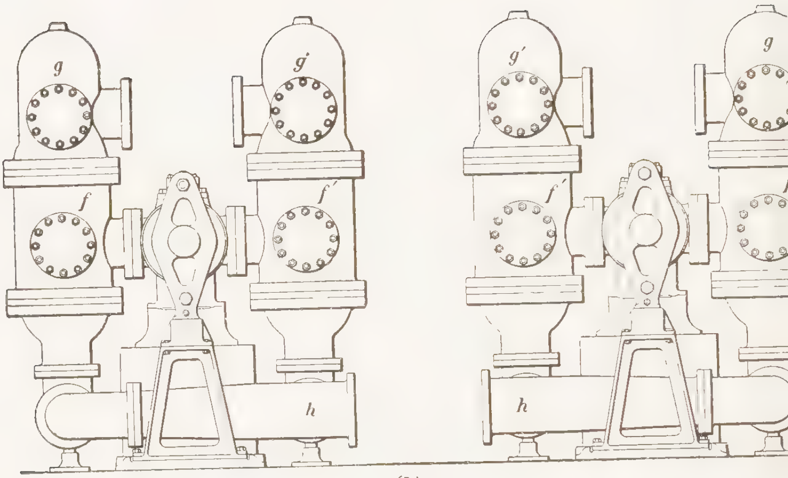
78. Fig. 32 (*a*) is a side view of the high-pressure side of a duplex pump driven by a cross-compound Corliss engine, the pump being of the double-plunger type. Fig. 32 (*b*) is an end view of the water end of both pumps, looking towards the engine, and Fig. 32 (*c*) is an end view of the engine, looking towards the flywheel, the observer being supposed to stand between the pumps and the engine. The plungers *a* and *b* are connected by yokes *c*, *c* and rods *d*, *d* and are driven

directly by the piston rods of the high-pressure and low-pressure cylinders, which for this purpose are prolonged beyond the pistons and pass through the back cylinder heads.

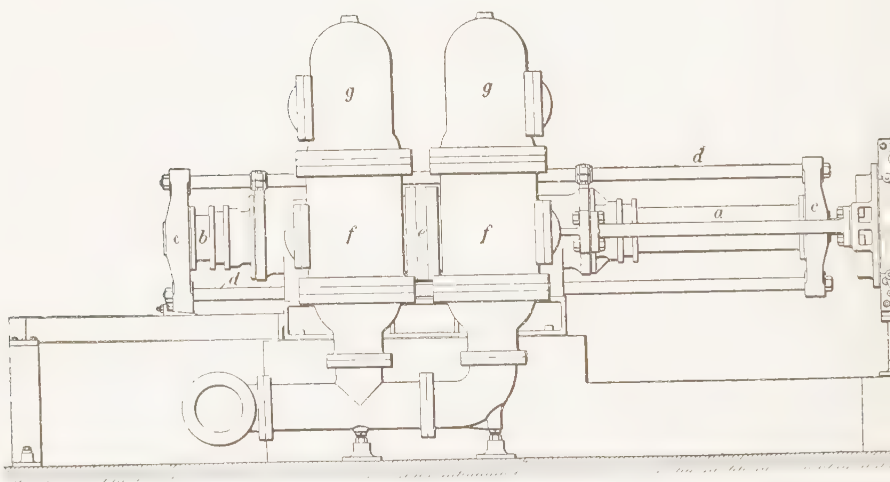
79. The pump cylinders have the necessary diaphragm *e* in the center, and each pump cylinder has two valve chambers *f*, *f'* containing the suction valves and two valve chambers *g*, *g'* containing the delivery valves. These valve chambers are placed on both sides of the pump cylinders. The four suction-valve chambers of each pump connect to the common suction branch *h*, and the two branches in turn are connected to the suction main by a **Y** fitting not shown in the illustration. The four delivery chambers of each pump are connected together by branch pipes, and these branch pipes in turn discharge into a common main delivery pipe.

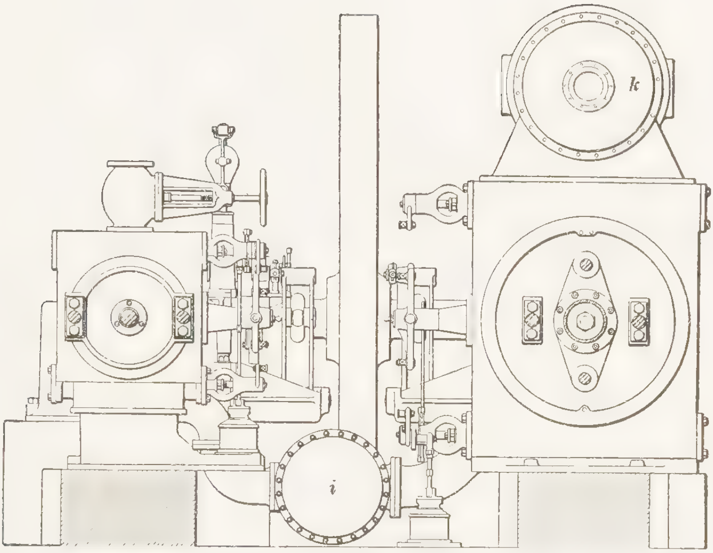
80. A reheating receiver *i* is placed between the high-pressure and low-pressure cylinders. The low-pressure steam-inlet valves are placed beneath the low-pressure cylinder; the low-pressure exhaust valves are on top and exhaust directly into the condenser *k*, which is placed on top of the low-pressure cylinder. The high-pressure valves are arranged in the usual way. The engine is provided with a variable speed Porter governor *l*, by means of which the speed of the engine may be varied to suit the requirements of the service.

81. The particular pump illustrated has cylinders 32 inches and 60 inches in diameter and a 48-inch stroke, the plungers being 13¾ inches diameter. It was designed by the Dickson Manufacturing Company, of Scranton, Pennsylvania, to pump water highly charged with sulphuric acid against a head of 700 feet. To guard against corrosion, the pump cylinders, valve chambers, and all pipes were lined with lead and the plungers and valves made of acid-resisting composition. Owing to the high economy possible through the use of a compound condensing Corliss engine, this is fitly called a "high-duty mine pump" by the builders.

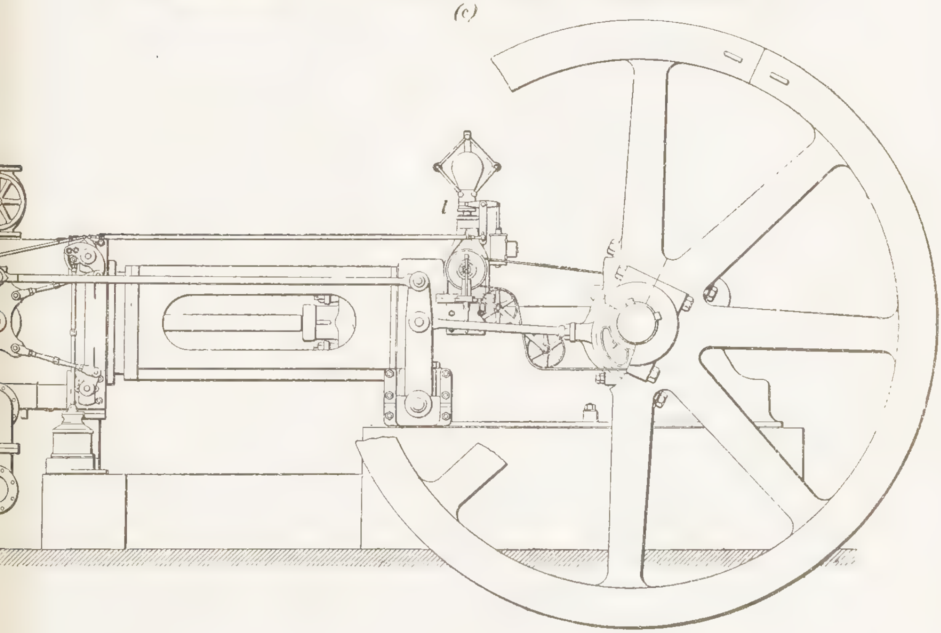


(b)





(c)



DISPLACEMENT PUMPS.

DEFINITION AND CLASSIFICATION.

82. A **displacement pump** is a pump in which there is a complete absence of moving parts and where the fluid to be pumped is moved by steam or compressed air. Of the steam-operated displacement pumps, the best known is the *pulsometer;* the *Harris compressed-air direct-air-pressure pump* and the *Pohlé air lift* are the best known air-operated displacement pumps.

THE PULSOMETER.

83. Fig. 33 shows a perspective view and Fig. 34 a sectional view of a pulsometer of the latest manufacture. In the sectional view the full lines represent the left-hand half and the dotted lines indicate the position of the discharge valves in the right-hand half of the pulsometer shown in Fig. 33. In the following description, the letters refer to both figures: The steam pipe is connected at *E* and the suction pipe at *S*. *C* is an air chamber that has no connection with *B* and *A*, but communicates with the suction pipe by means of the opening *I* situated below the suction valves *F* and *G*. The two latter valves are made of flat rubber and are held to their seats, as shown in the figure, by means of the spindles *R* and *T*. The spindles are raised and lowered, as the case may require, by means of the bolts *f* and *e*. *H*, *H* are plates that may be removed to facilitate the examination of the valves. *D* is a hard-rubber ball that acts as a valve for admitting the steam to the chambers *A* and *B*. *M* and *N* are exhaust valves, also made of rubber and situated in the chamber *L* attached to the other half of the cylinder. They are raised and lowered in the same manner as the suction valves by turning the bolts *g* and *h*. *K* is the delivery or column pipe.

84. The action of the pulsometer is as follows: Both chambers A and B are filled with water to about the height of the water in B, Fig. 34. The valve d is then opened and the steam enters one of the two chambers A and B. Sup-

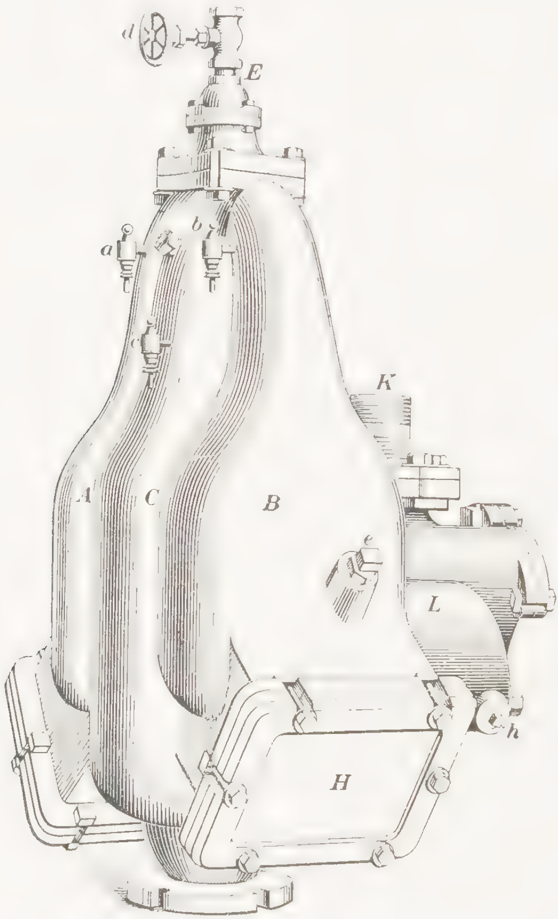


Fig. 33

pose it enters B, the valve D being at the right, as shown. The water in B will be forced through the delivery valve N into and up the column pipe K. This will continue until the water level gets below the edge of the discharge

opening P. At this point the steam and water mix in the discharge passage and the steam is condensed, creating a vacuum in B. The pressure in A is now greater than that in B, owing to the vacuum in B, and the ball valve D is shifted to the left, the steam entering the chamber A and

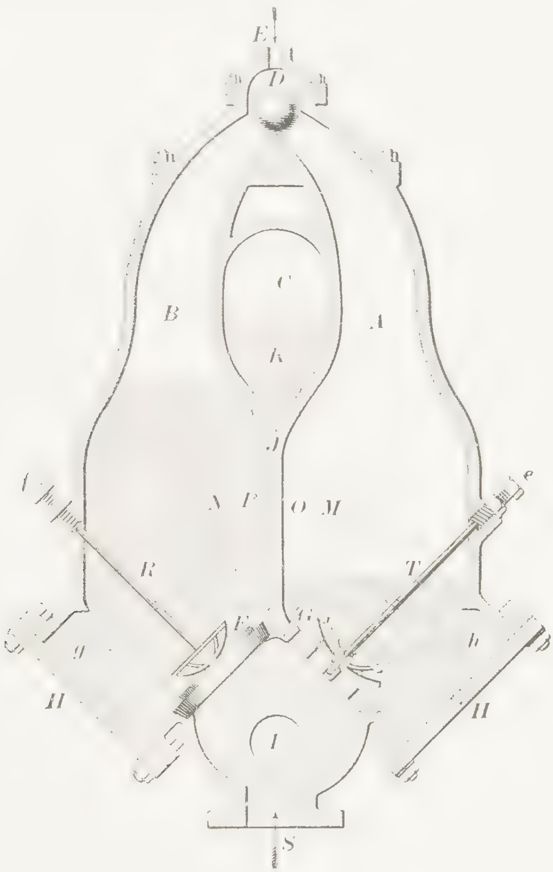


FIG. 34

driving the water through M into the passage O and column pipe K in the manner just described. While this is being done, the pressure of the atmosphere forces the water up the suction pipe S, opening the suction valve F, and into

the chamber *B*, filling it. When the suction valve is closed, owing to the reshifting of the ball valve *D* to the other side, the suction water enters the air chamber *C* through the inlet *I* and is brought gradually to rest by the compression of the air in *C*, thus preventing a shock owing to the sudden stoppage of the inflowing water. When the water in *A* has reached the level shown, the steam in *A* is condensed, the ball *D* is shifted to the right, and *B* becomes the driving chamber.

85. In Fig. 33 are shown three small air valves *a*, *b*, and *c*. The valve *c* admits air to the air chamber *C*, to replenish that which is lost through leakage and through absorption by the water. The valves *a* and *b* admit a small quantity of air to the chambers *A* and *B*, respectively, just before the suction begins. This injures the suction somewhat, but is necessary for two reasons: First, it acts as a regulator, governing the amount of water admitted to the chambers; and, second, it prevents the steam from condensing before the water gets below the edge of the discharge outlet. These valves open inwards, as before stated. Suppose there is a vacuum in *A* owing to the condensation of the steam. The atmospheric pressure forces open the valve *a* and admits a little air to the cylinder. The incoming water compresses this air and soon closes the valve. When the air has been compressed to such an extent as to balance the outside pressure of the atmosphere, the suction valve *G* will close and no more water can get in. Since the same thing occurs in the other chamber, it is evident that the amount of air admitted controls the amount of water admitted during the suction period, more water entering when there is less air in the chamber and vice versa. The admission of the air is controlled by turning the valves *a* and *b*, and these can be so adjusted that the suction valve in either chamber will close at the instant the ball is shifted to the other side, admitting the steam.

Moreover, the air prevents the steam from coming in contact with the water during the forcing process, until the

water level has sunk below the edge of the discharge orifice. Air being a poor conductor of heat, the steam does not condense until the mixture of the steam and water has taken place.

86. When the barometer stands at 30 inches, the pulsometer will raise water by suction to a height of about 26 feet, although it is not advisable to exceed 20 feet, and force it, when necessary, to a height of 100 feet. It has no wearing parts whatever except the valves, which are easily and cheaply repaired. It will work in almost any position, and when once started requires no further attention. There are no parts that can get out of order. It will pump anything, including mud, gravel, etc., that can get past the valves. Its first cost is low and it requires no foundations to set up. There is no exhaust steam to make trouble and no noise.

THE DIRECT-AIR-PRESSURE PUMP.

87. The direct-air-pressure pump here shown is the design of Professor Elmo G. Harris and is one of the simplest forms of pump. The pump is shown in diagrammatic form in Fig. 35. There are two pump tanks *a* and *b*, which are fitted with suction valves *c* and *d* and discharge valves *e* and *f*. The two tanks are connected to the common suction pipe *g* and both discharge into the same discharge pipe *h*. The tops of the pump tanks are connected by pipes *i* and *k* to an air compressor *m*, and by means of an automatically operated four-way cock *l*, either tank can be connected to the compressor side of the air compressor. The operation is as follows: with the cock *l* in the position shown, the tank *b* is connected to the suction side of the air compressor, and hence a vacuum is formed in the tank *b*. Consequently, the water in the supply is forced by atmospheric pressure up the suction pipe *g*, lifts the valve *d*, and passes into the tank *b*. At the same time the tank *a* is connected to the compressor side, and the air pressure on top of the water forces it out, the water holding the suction valve *c*

closed but opening the delivery valve e and passing up the discharge pipe h. When the tank a is nearly empty, the tank b

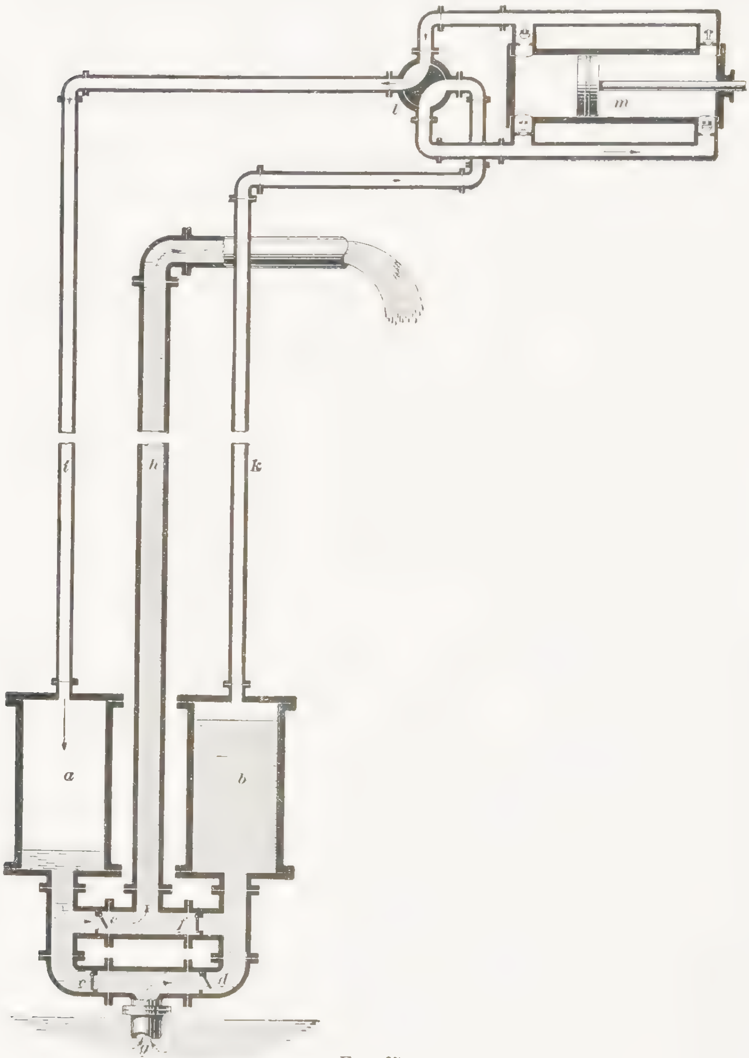


FIG. 35.

is nearly full; the cock l is then turned automatically so as to bring the tank a in communication with the suction side

of the air compressor and the tank *b* in communication with the compressor side. The water now flows into *a* and out of *b*, and the cycle of operations is repeated as long as the air compressor is working. The height to which water can be forced obviously depends on the pressure to which the air is compressed.

THE POHLÉ AIR LIFT.

88. The **Pohlé air lift** is much used for pumping water from artesian wells; it is operated by means of compressed air and has no moving parts. It is not affected by sand or

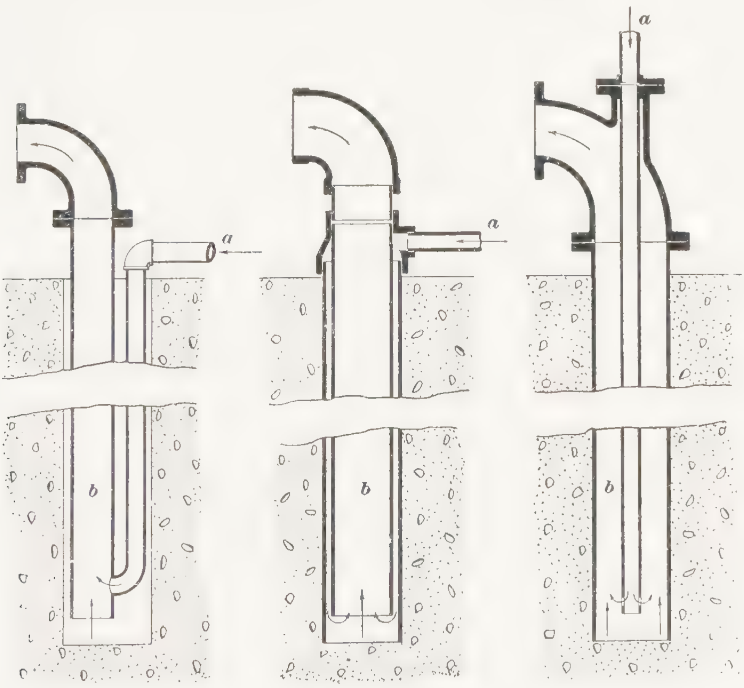


FIG. 36.

grit and the water is benefited to a considerable extent by the action of the air, in that it purifies and cools the water while it is being pumped. Other advantages claimed for

this device is that it increases the yield of an artesian well from two to five times; also, the full area of the well is available for a flow of water. Compressed air is supplied by means of an air compressor at the surface, which may be located in any convenient position, or one air compressor may supply several artesian wells.

89. The operation of the pump is as follows: Two properly proportioned pipes are inserted in the well, using either of the three arrangements shown in Fig. 36. Compressed air is supplied through the pipe *a* to the bottom of the well tube *b*. At the beginning of the operation the water inside and outside of the pipe is at the same level. When air is forced in through the pipe *a*, it forms alternate layers with the water, so that the pressure per square inch of the column thus made up of air and water inside of the water pipe is less than the pressure per square inch outside the pipe. This difference of pressure causes a continuous flow from the outside to the inside of the water pipe, and its ascent is constant and is free from shock or noise of any kind. The strata of compressed air in their ascent prevent any slipping back of water. As each stratum progresses upwards to the spout, it expands on its way in proportion to the overlying weight of water, so that the pressure of the air gradually becomes less and finally reaches the atmospheric pressure.

WATER ENDS OF RECIPROCATING PUMPS.

TYPES OF WATER ENDS.

90. Reciprocating pumps are either single-acting or double-acting. Single-acting pumps are either lift pumps, one of which is shown in Fig. 25, or outside-packed plunger pumps with one plunger, as shown in Fig 26, or outside-packed double-plunger pumps, as shown in Figs. 29, 30, and 32. Double-acting pumps are force pumps of the piston

or plunger pattern. Piston pumps, by reason of their construction, are inside-packed, and such a pump is shown in Fig. 6. Double-acting plunger pumps are inside-packed or center-packed. Attention is here called to the fact that outside-packed double-plunger pumps are often, but erroneously, considered as double-acting. While they give a discharge equal to that of a double-acting plunger pump, it is obtained by combining two single-acting plunger pumps to discharge into the same delivery pipe, and hence it is incorrect to call such a pump a double-acting pump. They are properly called **duplex pumps.**

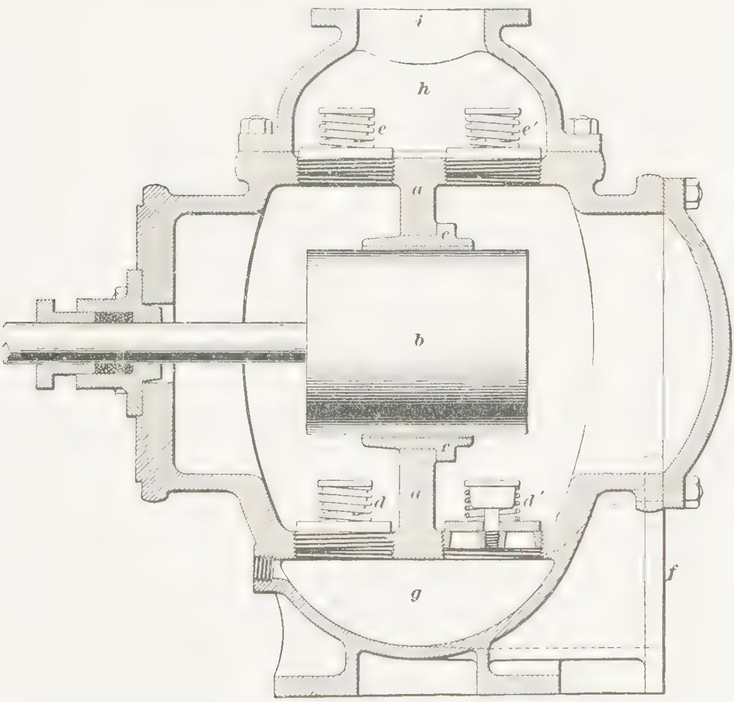


FIG. 37.

91. Fig. 37 shows the water end of a **double-acting inside-packed plunger pump.** The pump chamber is divided into two parts by a partition *a*, through which the

plunger *b* works back and forth. A water-tight joint between the plunger and partition is made either by a closely fitting

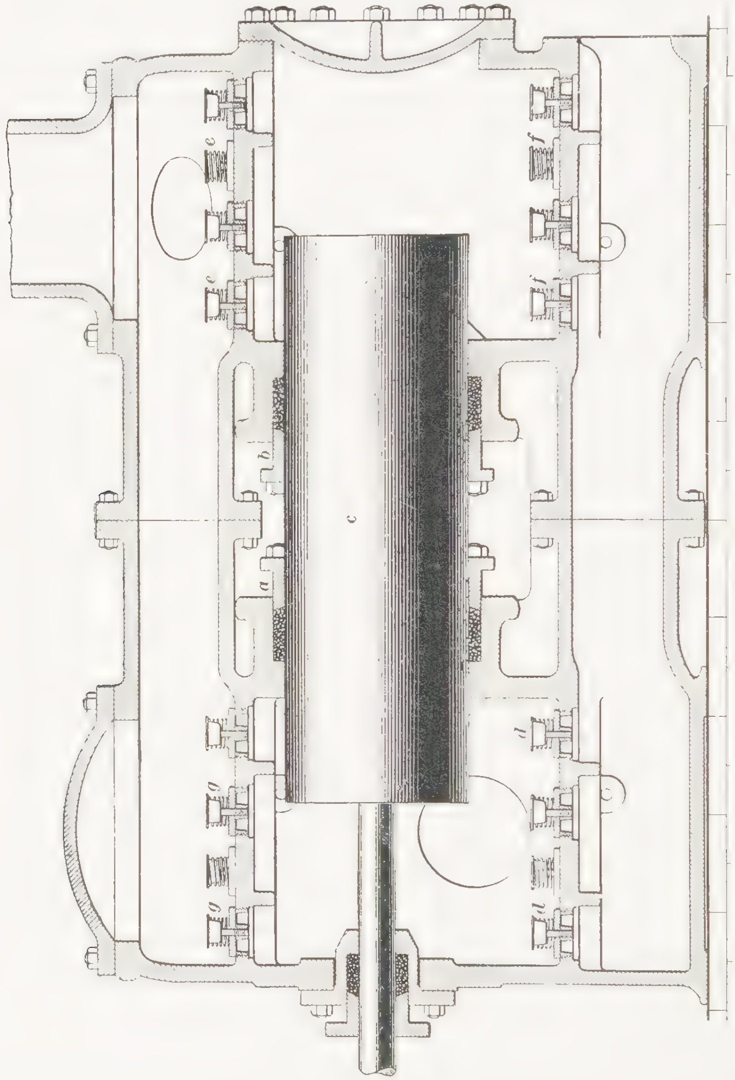


FIG. 38

bronze-lined bushing *c* or a regular stuffingbox and gland and fibrous packing. On either side of the partition is a set

of suction valves *d*, *d'* and delivery valves *e*, *e'*. The water enters the pump through the suction pipe, which is connected at *f* and flows into the suction-valve chamber *g*, from whence it passes to either side of the partition *a* and then into the delivery-valve chamber *h* and into the delivery pipe connected at *i*. When the plunger moves to the right, it displaces the water on the right of the partition *a*; the suction valve *d'* is closed by the pressure existing there, while the delivery valve *e'* is open and the water discharges into *h*. At the same time the plunger creates a partial vacuum at the left of the partition *a* and, hence, water flows through the open suction valve *d* into the left pump chamber. The delivery valve *e* is kept closed by the pressure in *h*. When the plunger moves to the left, the suction valve *d'* and delivery valve *e* open and the suction valve *d* and delivery valve *e'* close. It is thus seen that the pump discharges during either stroke of the plunger, i. e., the pump is double-acting.

92. Fig. 38 shows a sectional view of the water end of a center-packed double-acting plunger pump, the stuffing-boxes *a* and *b* being used for packing the plunger *c*. The action of the pump is identical with that of the pump shown in Fig. 37, that is, when the plunger moves to the right the suction valves *d*, *d* and delivery valves *e*, *e* are open and the suction valves *f*, *f* and delivery valves *g*, *g* are closed. When the plunger moves to the left, the suction valves *f*, *f* and delivery valves *g*, *g* are open and the suction valves *d*, *d* and delivery valves *e*, *e* are closed.

93. The water end of a double-plunger pump for high pressures is shown in Fig. 39. The two plungers *a*, *b*, as usual, are connected by yokes and side rods outside of the pump. The rods *i*, *i* tie the water end to the steam end. Each plunger has its own suction valve *e* and delivery valve *f*. The suction valves communicate with a common suction chamber, to which the suction pipe *c* is attached. At *d* the discharge pipe is shown. Plugs *g*, *g* when removed give access to the valves. A standard *h* supports the water end

on its foundation. The illustration clearly shows that each plunger is single-acting, but that the discharge is equal to that of a double-acting pump. Pressure pumps do not differ

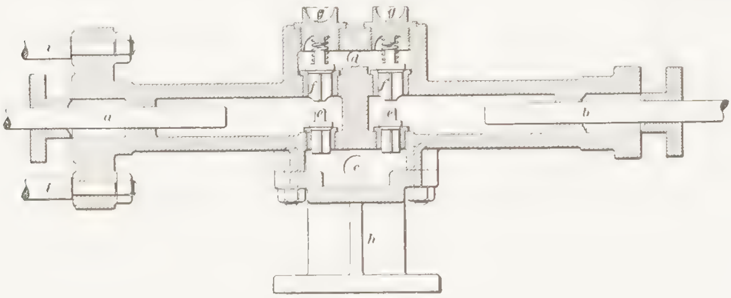


FIG. 39.

in their operation from ordinary pumps; all parts are simply made extra heavy so as to stand the high pressure, and for very high pressures steel is substituted for cast iron in the water end.

94. Fig. 40 shows in diagrammatic form two forms of a plunger pump that is double-acting and is known as a **differential pump.** Its distinguishing feature is that it needs only one set of suction valves and delivery valves. Fig. 40 (*a*) shows the arrangement used for two plungers *a* and *b*, which are connected together by yokes and side rods. In Fig. 40 (*b*), the two plungers are connected directly together. In both designs one plunger, as *a*, has exactly double the area of the other plunger *b*. This fact must be carefully borne in mind. Since the stroke of both plungers is the same, it follows that the larger plunger in Fig. 40 (*a*) will displace double the quantity of water that the smaller plunger displaces. In Fig. 40 (*b*), the left-hand side of the plunger *a* displaces double the quantity of water displaced by the plunger *b*. In both designs *c* is the suction valve and *d* the delivery valve.

95. The operation of the differential pump shown in Fig. 40 (*a*) is as follows: The pump being filled with water and the plungers moving to the right, the suction valve is

open and the delivery valve closed. The plunger *b*, or the right-hand side of the plunger *a* in Fig. 40 (*b*), forces a volume of water equal to its displacement out of the chamber *e* and up the delivery pipe *f*. At the same time, double the volume of water is drawn into the suction chamber *h*.

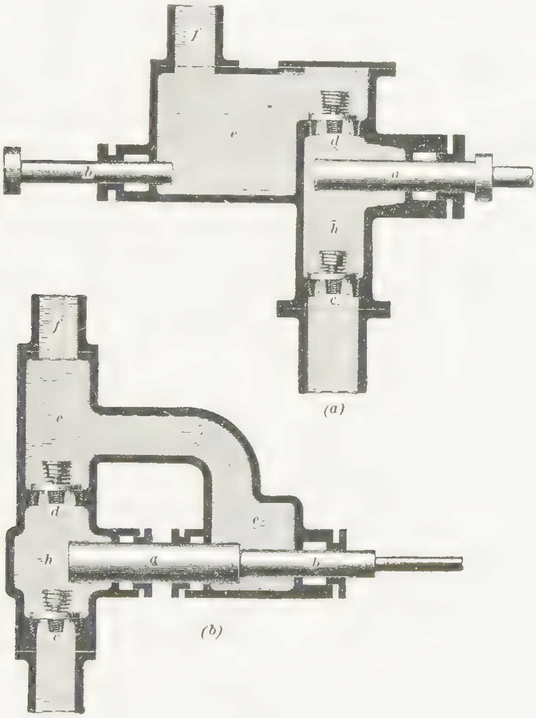


FIG. 40.

Now, assume that the plungers move to the left. The suction valve is then closed and the delivery valve is open, and double the quantity of water discharged during the stroke to the right now flows into the chamber *c*. But while this is going on, the volume of the chamber *e* increases by the receding of the plunger *b*, or the outward movement of the plunger *a* in Fig. 40 (*b*), by an amount that at the end of the stroke is equal to exactly one-half the amount discharged into it, so that the outflow into the delivery pipe is

only one-half of that discharged into the chamber *e*. This outflow is equal to the displacement of the small plunger, or the right-hand end of the plunger *a* in Fig. 40 (*b*), and hence the same amount of water is discharged during both strokes.

RIEDLER PUMPS.

96. Development.—Riedler pumps are the invention of Professor Riedler and are a type of pump designed for running at very high speeds. By study, experimenting, and careful noting of cause and effect, he discovered several very important phenomena. He found that there was much greater resistance to the flow of water through the valve passages and ordinary pumps than was before this thought to exist. He further found that the slip of ordinary valves is very large, and that even when small has a great tendency to cause severe hydraulic shocks throughout the pressure parts of the pump. He also was aware that the frictional resistance to the passage of a certain quantity of water through a large number of small openings is much greater than that existing when the same quantity of water passes through a single opening equal to the combined area of the smaller ones. With these facts in view, Professor Riedler designed a pump valve having the useful valve area as large as possible and containing as few separate passages as is consistent with good construction. He substituted one large valve for many small ones, thus decreasing the friction of the water in the valve passages. The reduction of the slip was accomplished by arranging a mechanical controlling device, whereby at the proper time and without restricting the water passage the valve was closed. The mechanical controlling device further assists in the reduction of friction in the valve passages, as it permits the valve lift to be high, thus increasing the effective area.

97. The first pumps fitted with Riedler valves were constructed in 1884, since which time more than 1,500 pumps have been built. These pumps are adapted to any service

to which pumping machinery may be applied. They are built in all sizes, ranging in capacity from 115,000 gallons in 24 hours to 20,000,000 gallons in 24 hours, and are working under heads as high as 2,480 feet and at speeds as high as 120 revolutions per minute, and with piston speeds as high as 606 feet per minute, which, by the way, is the average speed of steam pistons.

98. Valve Gear.—Fig. 41 shows an outside view of a direct-connected, electrically driven, differential Riedler pump having the plunger arrangement shown in Fig. 40 (*b*). The pump valves are closed by cranks, the crank *a* operating the suction valve and the crank *b* the delivery valve. The two cranks are operated from a wristplate *c* similar to that of a Corliss engine and to which they are connected by the rods shown. The wristplate is rocked back and forth by the eccentric *d* on the crank-shaft, to which it is connected by the eccentric rod *e*. The plungers are driven by a crank, as shown.

99. Riedler Valve.—Fig. 42 shows a detail of the improved Riedler suction and delivery valve. Both suction and delivery valves are alike in these pumps except as regards the flange for securing them to the pump chambers. The valve proper consists of three concentric bronze rings *a*, *b*, and *c*, each of which is cast in one piece and which are set into a spider *d* having eight arms. This spider is free to move up and down on the central valve post, or valve spindle *e*. This valve rests on a heavy cast-steel valve seat *f* having three annular openings *a'*, *b'*, and *c'*. The valves proper are not rigidly connected to the spider, but each valve is free to form its seat with the valve seat and independent of the spider or each other. A leather ring between the valve proper and the spider serves to make an absolutely tight joint. A circular nut *g* is secured to the top of the hub of the valve spider *d* and holds in place a steel pressure plate *h*. This pressure plate rests on top of a spring cap *i*, below which a spring *k* of soft rubber is placed. This rubber allows of a certain amount of yield between the valves

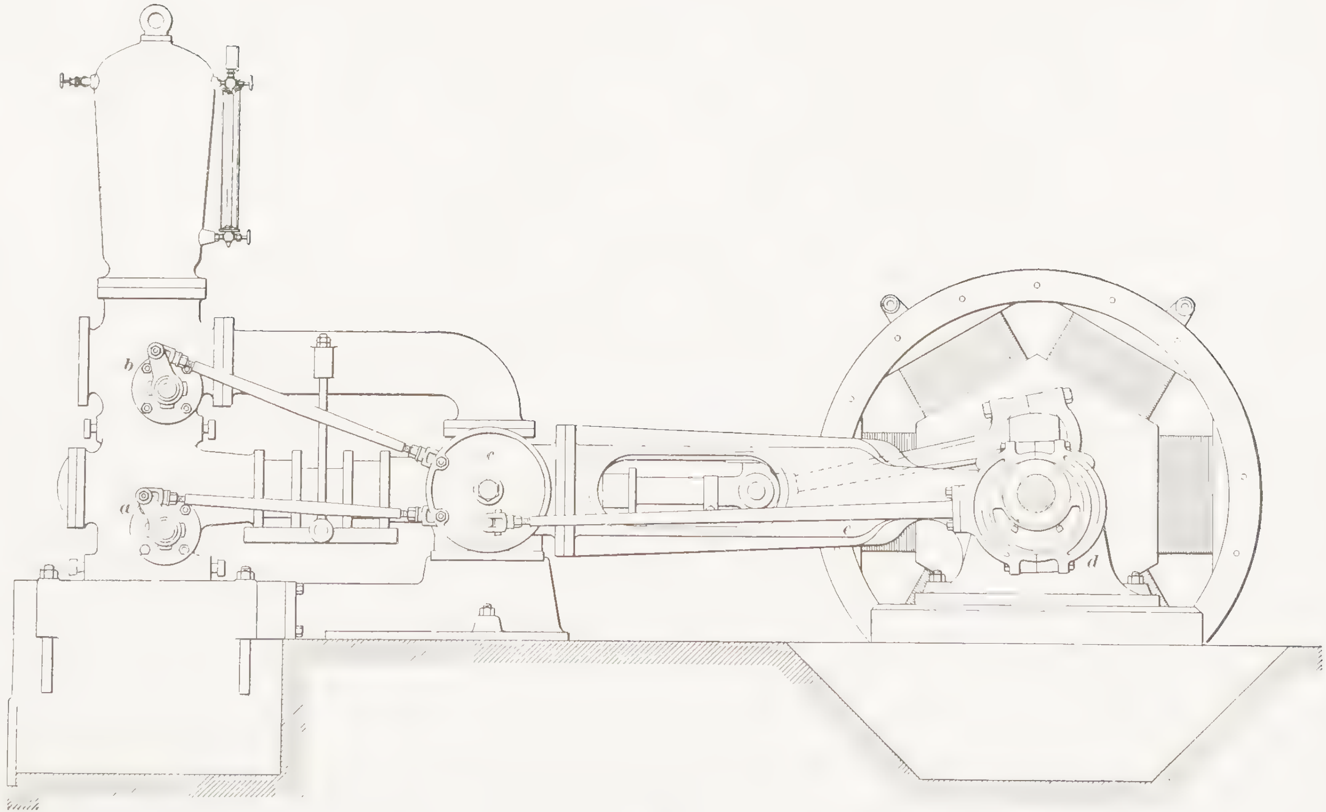


Fig. 11.

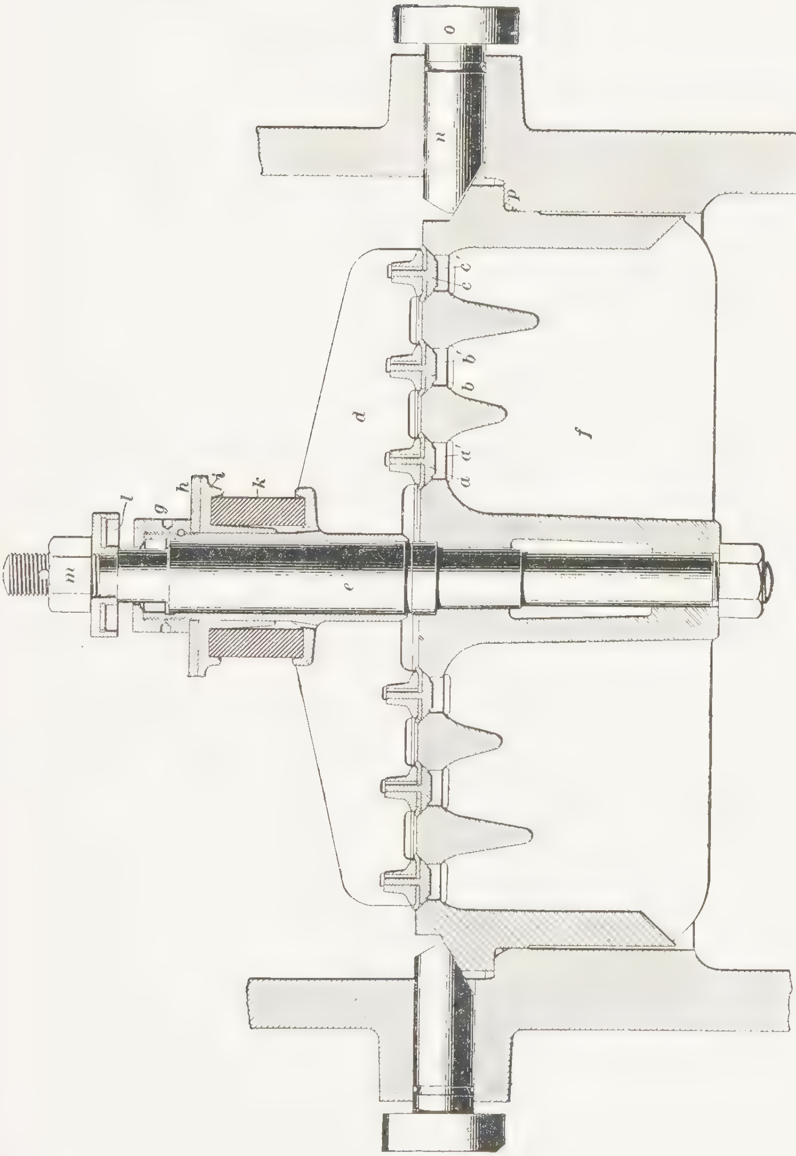


FIG. 42.

and its seat in case any foreign matter should get between them. Two steel fingers, not shown in the drawing, press upon the pressure plate and serve to close the valve just before the piston reaches the end of its stroke. A water cushion *l*, the object of which is to prevent the valve from striking its stop when opening, is secured to the top of the spindle *e* by the nut *m*. The nut *g* is closely fitted to the chamber in *l* and traps the water in front of it, thus making a hydraulic cushion. The valve seats are secured in the valve chambers by wedge-shaped plugs *n*, *n*, which are forced in by studs and nuts through the gland *o*, the effect being to force the valve seat *f* hard down on its bearing *p* in the pump chamber.

100. Fig. 43 is a perspective view of the Riedler valve and seat, showing the operating mechanism by means of which the valve is seated. All visible parts are lettered the

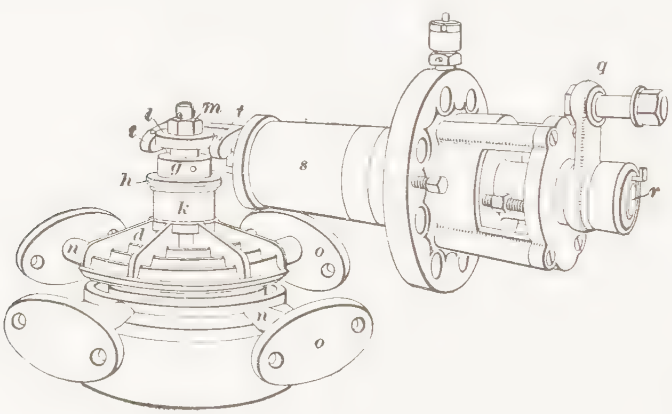


FIG. 43.

same as in Fig. 42. The crank *q* is operated from the wristplate shown in Fig. 41; it is keyed to a shaft *r*, which passes through a stuffingbox *s* bolted to the valve chamber and carries a forked crank at its inner end. The jaws or fingers *t*, *t* of the forked crank press upon the pressure plate *h* to seat the valve at the proper time. The motion of the fingers is so timed in relation to the motion of the

plungers that the fingers are clear of the pressure plate h when the plungers begin to deliver water, thus leaving the valve free to open.

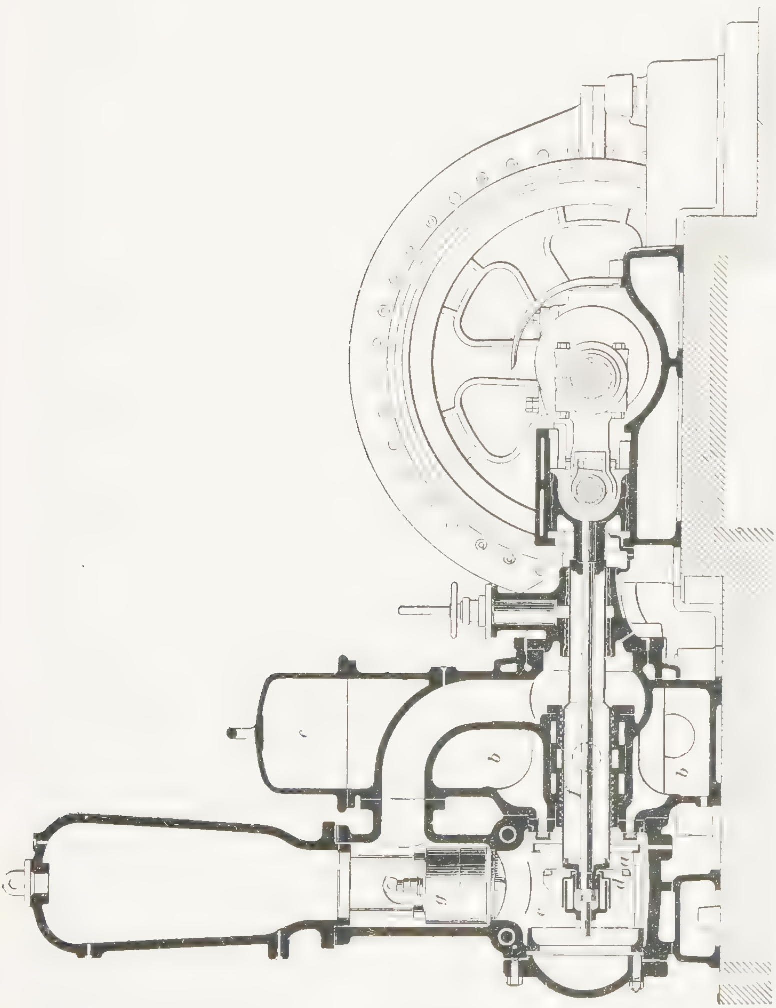


Fig. 44.

101. The Riedler valve is by no means confined only to water pumps. It has been and is used successfully for high-pressure air and gas compressors. The Riedler pump may

be driven by a steam engine, electric motor, turbine water-wheel, by belting, or in any other convenient manner.

102. Riedler Express Pump.—A type of Riedler pump that has recently been brought out for running at a very high speed is called the **Riedler express pump** and is shown in Fig. 44. Although the ordinary Riedler pump can be run at speeds as high as 150 revolutions per minute and sometimes faster, conditions arise requiring a much higher speed, and to meet this condition this special design, which may be run at speeds as high as 300 revolutions per minute, has been developed by Professor Riedler. The main feature of this pump—in fact, the part that permits running at such high speeds, is its suction valve. As will be seen by referring to the figure, the suction valve *a* is annular in form and is concentric with the plunger; it lifts in the direction opposite to that of the plunger when on its suction stroke, the water flowing from the suction chamber *b b* into the valve chamber *c*. At the end of the suction stroke a buffer *d* mounted upon the end of the plunger drives the suction valve to its seat, making it certain that the valve is seated when the plunger starts on its delivery stroke and allowing practically no slip. A high suction air chamber *e*, containing a column of water, is placed above the suction valve, making it certain that the pump will fill as the plunger *f* makes its suction stroke. The delivery valve is shown at *g*. It will be noticed that this pump is of the differential type.

103. The chief point of advantage of the express pump is that it may be connected to high-speed motors. It is of small dimensions compared to the quantity of water it can handle, and thus consequently low in first cost. About thirty of these pumps have been constructed up to the year 1901, ranging in capacity from 152,000 gallons in 24 hours to 7,600,000 gallons in 24 hours, and in speed as high as 300 revolutions per minute, pumping against a head of 820 feet; others have been built to pump against a head of 1,800 feet at 200 revolutions per minute.

PUMPS.

(PART 2.)

DETAILS OF PUMP WATER ENDS.

PUMP PLUNGERS.

CONSTRUCTION.

1. The smaller sizes of pump plungers are usually made of solid round bars of metal turned smooth, so as to work through a stuffingbox with as little friction and wear as possible. For larger sizes the plungers are frequently of cast iron and are often made hollow to reduce the weight and amount of material required. Incidentally, it may be remarked that a hollow plunger is easier to move than a solid one, all other conditions being equal. This is due to the fact that the water buoys up a hollow plunger more than a solid one. In large horizontal pumps hollow plungers are often so proportioned that they actually float in the water, thus relieving the stuffingboxes of the weight of the plungers and reducing the wear.

2. Fig. 1 shows a simple form of solid plunger pump, such as is often used for feeding boilers. The plunger works through a stuffingbox of the ordinary pattern, packed with hemp or some of the common types of soft piston-rod packing.

3. Fig. 2 shows three styles of large, hollow, cast-iron

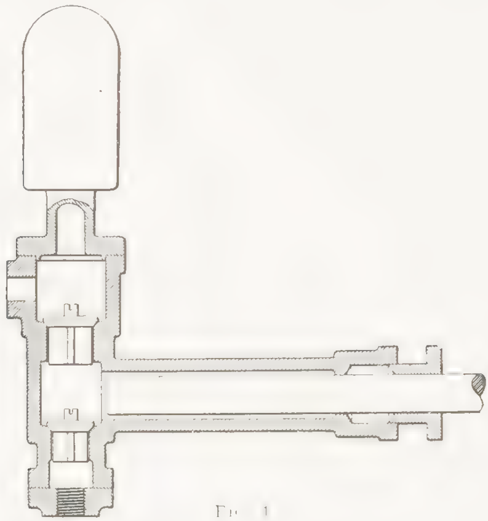


Fig. 1

plungers, with methods of attaching them to the pump rods. The packing for these plungers, when used for moderate pressures, is usually hemp contained in a stuffingbox of the ordinary pattern.

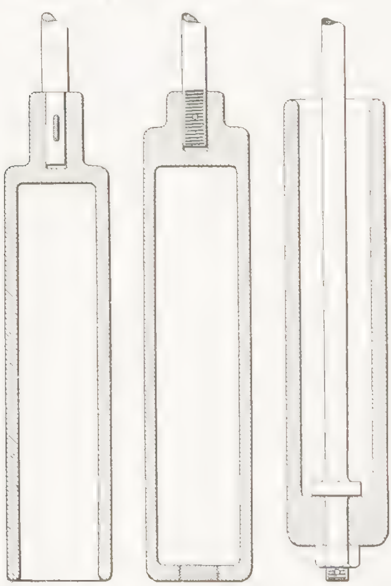


Fig. 2.

PLUNGER PACKING.

4. When the pressure under which the pump works is very heavy, U-shaped leather packing is sometimes used. Fig. 3 shows three methods of holding these **cup leathers,** as they are called. The section at (*b*) shows the leather *o* held in a recess cast in the upper end of the

pump cylinder. In this case it is necessary to remove the plunger *D* in order to insert a new leather or to examine an old one. Experience also shows that the leather bears against the plunger with the greatest force at the bend *B* and fails at that point first. In (*c*) the leather is held in its recess by a gland *s*, and is also supported by a brass ring *C*,

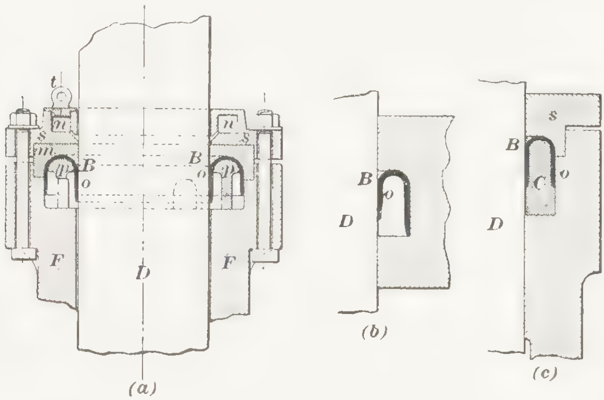


FIG. 3.

which prevents the severe pressure of the leather against the plunger at *B*. A more elaborate packing is shown at (*a*); the gland *s* is lined with a brass ring *m*, which holds the leather *o* down on a brass supporting ring *p*. A chamber *n* in the gland serves to hold oil for lubricating the plunger.

The form of packing shown at (*b*) is cheap, but in addition to the difficulty of inserting the leather, it is difficult to cast the recess so that it will fit the leather properly. In either of the forms shown in (*a*) and (*c*), the gland can be accurately turned to bear against the curved portion of the leather, thus forming a better support and increasing the life of the packing.

5. Fig. 4 shows an inside-packed plunger with a removable stuffingbox designed for hemp packing. This construction is better than merely providing a close-fitting bushing, especially when the water is gritty and thus liable to wear the plunger

Inside-packed plunger pumps have several disadvantages. When the packing becomes worn, the heads of the pump cylinder must be removed in order to tighten or renew it, and,

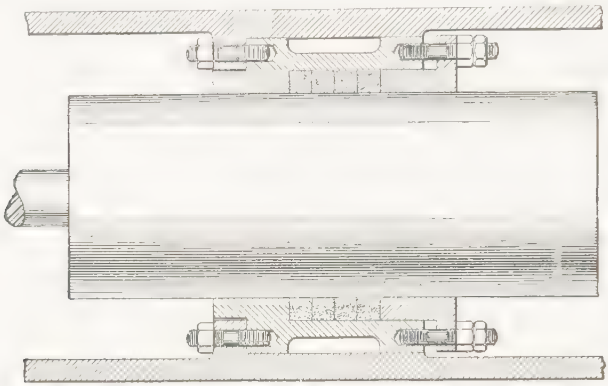


Fig. 4.

besides, there is no way of detecting leakage when the pump is working. With gritty water, especially when working under high pressures, these disadvantages become serious.

6. Fig. 5 shows a good arrangement of plunger, stuffingbox, and gland. This type of plunger and stuffingbox is

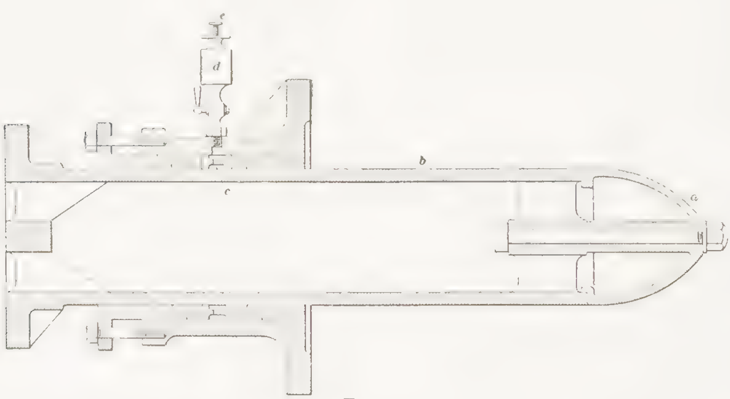


Fig. 5.

much used in mining pumps. The plunger cap *a* is made of acid-resisting metal, while the plunger *b* proper is made of cast iron, it having been found in mining work that the

plunger cap or point is the only part that is attacked by acid water. Apparently the play of the plunger through the stuffingbox and grease prevents the water attacking its surface. An improved form of **grease ring** is shown at *c*. This ring fits into the stuffingbox and is placed between the rings of fibrous packing. It is recessed both inside and outside and has several holes by which the outside recesses connect with the inside recesses. The outside recess is in connection with the grease cup *d*, which is provided with a cock. When it is desired to grease the plunger, the cock is opened and the grease forced in the space around the grease ring by the screw *e* on top of the grease cup. This is done once or twice during the day, and the cock is then closed so as to relieve the grease cup of the water pressure and to prevent consequent leakage. The stuffingbox is bolted directly to the pump chamber, which may be of any type, but for high-pressure mine work it is generally circular. This type of plunger and stuffingbox has been used with much success in the anthracite coal regions.

PUMP PISTONS.

7. Pistons for force pumps are made in a variety of forms. Fig. 6 shows a piston with fibrous packing held in place by a follower. The follower is fastened to the piston by means of an extension of the piston rod beyond the nut that holds the piston in place.

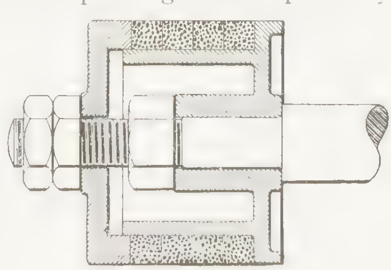


FIG. 6.

8. An excellent packing for small pistons is shown in Fig. 7. It consists of a metallic piston made up in three parts, between which are clamped two cup leathers, as shown.

9. Pistons for suction and lift pumps must be provided with valves that allow free passage for the water through

the piston in one direction and prevent its return. These

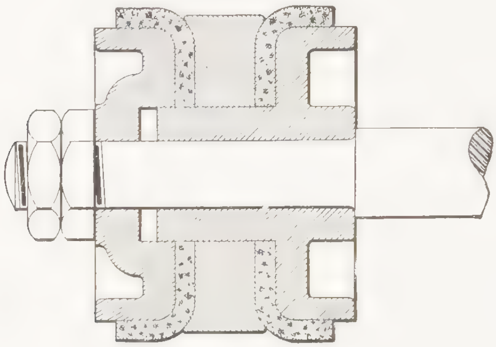


Fig. 7.

valves may be of any design that will furnish the required area of passage and at the same time will be strong enough to withstand the pressure of the water.

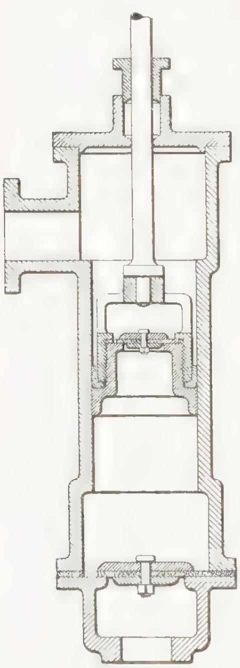


Fig. 8.

10. For small pumps and moderate lifts, leather **clack valves**, Fig. 8, are often used. They consist simply of a leather disk held at one side and strengthened by a metal plate on top. The leather when wet forms an excellent hinge and a tight valve. Leather clack valves are also used for the suction and delivery.

11. For lift pumps working under high pressures, the valves shown in Fig. 9 give good results. The piston shown at (*a*) has a rubber disk valve working on a gridiron seat. The valve is guided by a central spindle *s* and is held on its seat by a light helical spring that acts on a plate on top of the rubber disk. This piston is very long and has no separate packing.

12. The valve shown at (*b*) is for very heavy pressures It consists of a metal disk guided by a central spindle *s* and held down by a helical spring in the same manner as the

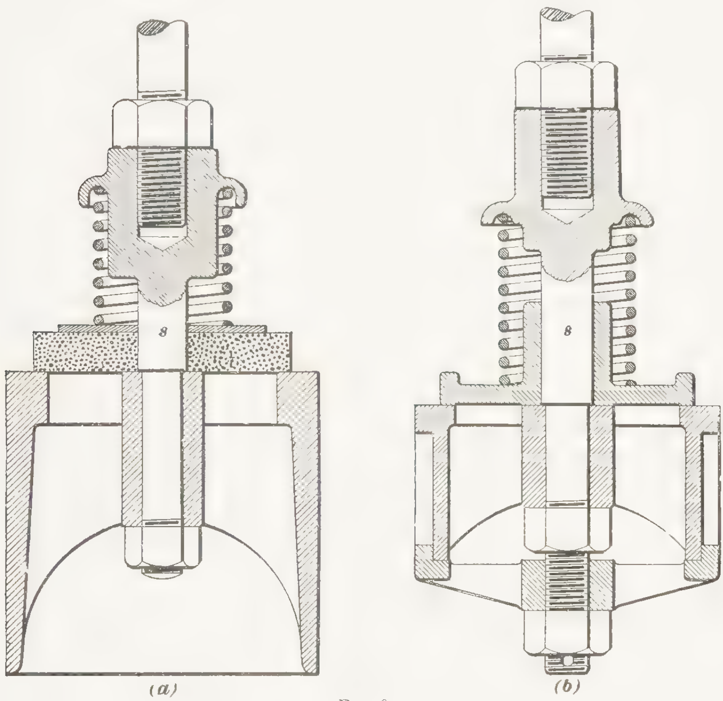


Fig. 9.

rubber valve. The piston is made with a follower plate for the purpose of holding a fibrous packing in the same manner as the piston shown in Fig. 6.

PUMP VALVES.

REQUIREMENTS.

13. The most important details of a pump of any kind are the valves. They must be so designed and constructed that they will fulfil all the following conditions as thoroughly as possible:

(*a*) They must open freely under a light pressure.

(*b*) The net area of the passages through the valves should be great enough to limit the velocity of flow through them to 240 feet per minute.

(*c*) The lift of the valves should be small.

(*d*) The passages for the water should be as direct as possible.

(*e*) The valves must close tightly under all conditions.

(*f*) The valves and their seats must be durable and of such materials as are not easily affected by the impurities in the water.

(*g*) The valves must return to their seats quickly and without shock as soon as the current through them is stopped.

(*h*) The valves and seats must be easily repaired or removed when worn.

A great variety of valves have been designed with a view of satisfying these requirements, taking into consideration the widely varying conditions under which pumps must work.

CONSTRUCTION.

14. Disk Valves. Fig. 10 shows two valves of a type much used in all classes of pumps for ordinary pressures and service. The valve *v* consists of a vulcanized India-rubber disk that rests on a gun-metal or brass seat *s*. The seat is threaded at *t*, so that it can be screwed into the deck of the valve chamber and thus can be easily removed. The part of the pump chamber that contains the valves is usually called the **valve deck,**

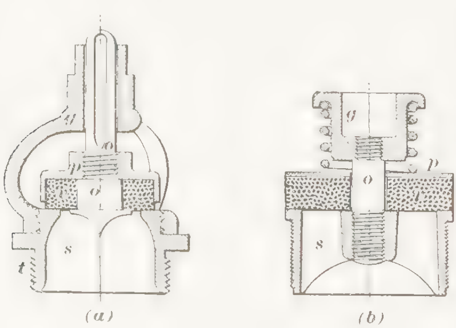


FIG. 10.

and it is spoken of as the **suction valve deck** and **delivery valve deck** in accordance with the kind of valves it carries. In the design shown at (*a*), the valve is fastened to a spindle *o* by a cap *p*. The spindle is guided by a cage-shaped guard *g* screwed on to the valve seat. The lower end of the spindle is made conical, so as to change the direction of motion of the water gradually and to reduce the resistance to flow. In the design shown at (*b*), the spindle *o* is screwed into the valve seat and carries a guard *g*. A helical spring between this guard and the plate *p* helps to seat the valve quickly.

The size of these valves varies from 2 to 6 inches in diameter, the most common size for ordinary conditions being 3 inches.

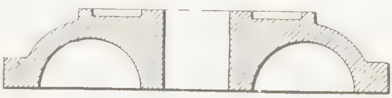


FIG. 11.

15. When used for pumping *hot* water, the disk must be made of a composition that will not be affected by the heat and for very high pressures metal disks are used, generally of the form shown in Fig. 11.

16. Fig. 12 shows the construction of a large disk valve, such as is often used in mine pumps. The valve seat *A* is held in place by the flange *B* and is perforated, as shown in the top view of the seat, by a large number of small holes. The valve *C* is made of soft rubber and is placed within the bronze or composition cap *D*. The head of the bolt *E* forms a stop and the spring *S* assists the valve in closing.

17. Clack Valves.—A section of a clack valve is shown in Fig. 13. The **clacks** *A* and *B* are lined with leather on the bottom so as to make a tight fit on the seat without having to do much fitting. A stop *C* prevents the valves opening too far, while *E* is the pin on which the clacks are hinged. A cylindrical casing *D* forms the valve seat; it may be easily renewed when worn. These valves are of the type known as the **butterfly valve,** and are much used for pit pumps at mines on account of their cheapness and simplicity of construction.

18. Single-Seat and Double-Seat Valves.—A **single-seat valve** that is suitable for high pressures, up to heads of 500 feet, is shown in Fig. 14, where *A* is the valve; *B* is

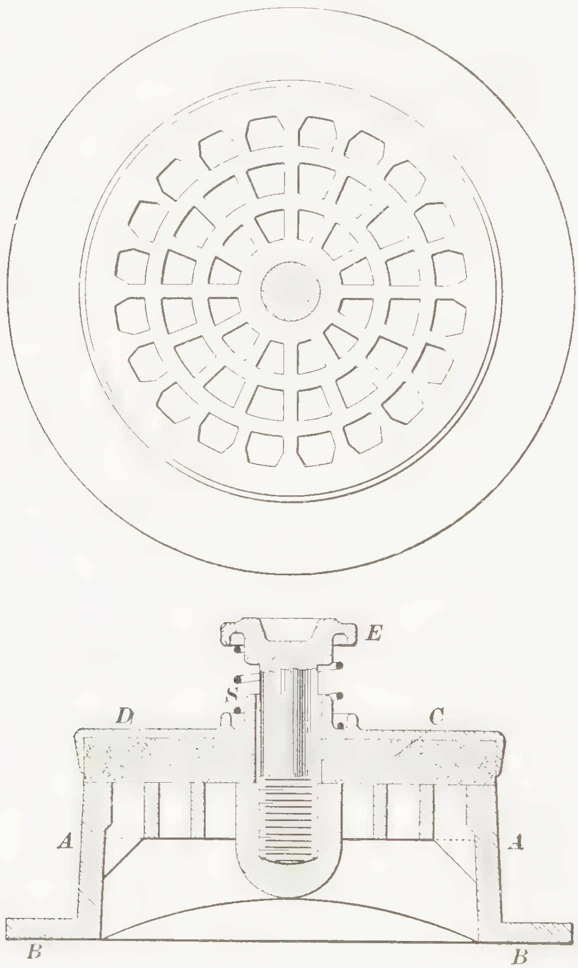


FIG. 12.

a stem solid with the valve that acts as a guide inside the bearing *D*; and *C*, *C*, *C*, *C* are rubber rings which are kept in position by means of the stem and are separated by the

washers *E*, *E*, *E*. These rings prevent shock as the valve lifts and also help to close it quickly, thus serving the same purpose as the helical spring in Fig. 10 (*b*).

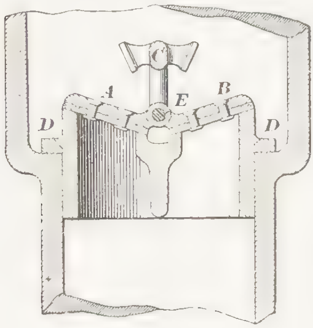


FIG. 13.

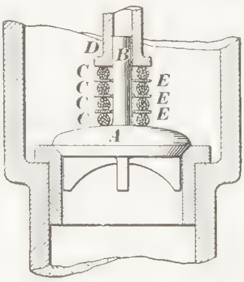


FIG. 14.

19. A section of a **Cornish double-seat valve** is shown in Fig. 15. This valve gives excellent results when used in

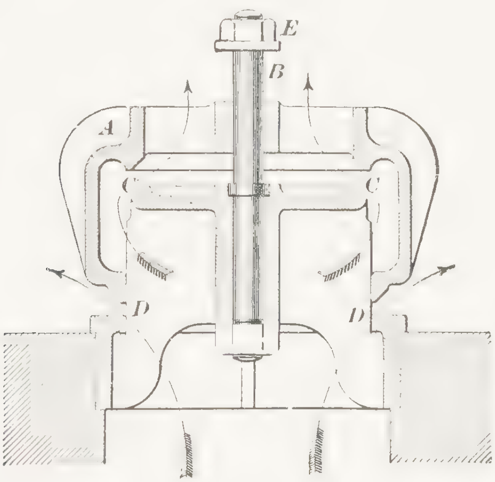
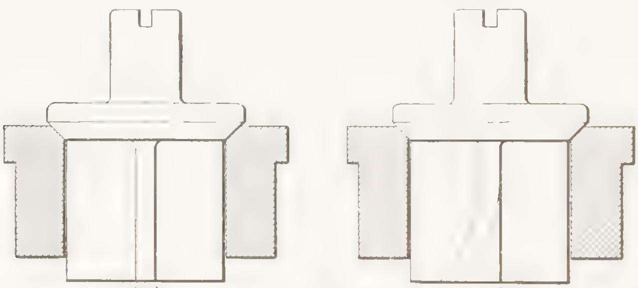


FIG. 15.

large pumps working under high pressures and has been applied to pumps working under heads up to 700 feet. It is called a *double-seat* valve because it has two seats and two

openings for discharge. The casing *A* slides on the vertical stem *B*, its lift being regulated by the nut and washer *E*; when down, it rests on the valve seats *C* and *D*. When the pressure below becomes greater than that above, it raises the casing, and the water is discharged through the circular openings at *C* and *D*. The rib around the outside of the casing is for the purpose of strengthening it. The valve seats are conical. The figure shows that one opening discharges the water under the lower edge of the valve and the other through the inside.

20. Wing Valves.—The **wing valve** shown in Fig. 16 (*a*) is largely used in power pumps for feeding boilers and in



(*a*) (*b*)

FIG. 16

hydraulic pumps for high pressures. The valve and seat are made either of hard brass or of gun metal and are ground together to secure tight closing. The lower portion of the wings is sometimes curved as shown at (*b*), the object being to give the valve a partial rotation at each stroke of the pump. This compels it to seat at a new place with each stroke and tends to wear the valve and seat more evenly.

21. Pot Valves.—Fig. 17 (*a*) is a sectional view of a **pot valve.** This type of valve is used principally on mining pumps for lifts up to 1,000 feet. They are made separate from the pump chambers and may be readily replaced when broken or worn. The cover *a* is secured by hinged bolts, so that it may be quickly removed for access to the valve *b* and the valve seat *c*, which is made of composition and pinched

between the pot and the pump chambers. The valve spring *d* surrounds the valve guide *c*.

22. Fig. 17 (*b*) shows a type of pot valve used for high lifts up to 1,200 feet. The valves are made small and faced with hard rubber; a group of them is placed in one heavy

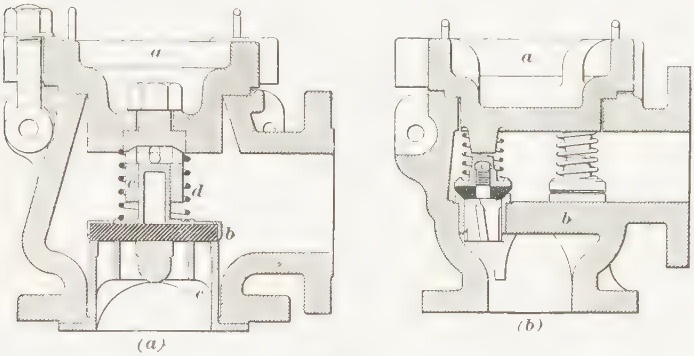


FIG. 17.

pot which is bolted to the pump chamber. Access to the valves may be had by removing the cover *a*. The valve seats are made of composition bushings forced into the valve deck *b*.

AIR CHAMBERS.

PURPOSE.

23. Even in double-acting pumps there is an interruption of the flow at the end of the stroke, when the piston changes its direction of motion. This has the effect of bringing the column of water in the suction and discharge pipes to rest at the end of each stroke, and this column of water must be set in motion again as the next stroke is made. If the pipes are long, the force required to stop and start the water will be very great, and there will be a severe shock at the end of every stroke that will absorb power and subject the pump and pipes to great stresses.

This difficulty is removed and the flow through the pipes is made more continuous and steady by the use of **air chambers.** An air chamber is a vessel containing air and is attached either to the pump just outside of the discharge valves or to the discharge pipe near the pump. While small duplex pumps are often run without an air chamber, it is better in general to fit one to all pumps, since its effect will always be beneficial.

DELIVERY AIR CHAMBERS.

24. Principle of Action.—Fig. 18, which shows an air chamber attached to the discharge pipe of a single-acting

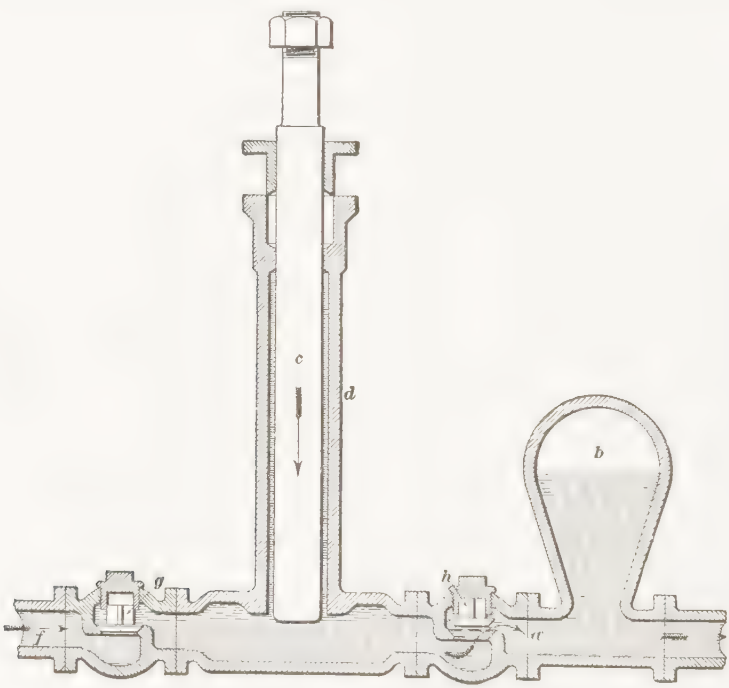


FIG. 18.

plunger pump *d* for boiler feeding, will illustrate the principle of action of an air chamber. The water, after being

drawn in through the pipe *f* past the valve *g*, is forced by the plunger *c* past the valve *h* into the discharge pipe *a*, part of it flowing into the air chamber *b* and compressing the air therein. When the plunger reaches the end of its stroke and no more water is being forced into the discharge pipe, the compressed air in the air chamber forces the extra water out through the discharge pipe. In this way the air chamber acts as a *reservoir* that receives its supply during the inward motion of the plunger and gives it out again in a nearly steady stream. The air in the air chamber acts as a spring that absorbs the extra force during the inward stroke of the plunger and gives it out during the return stroke, thus relieving the pump and pipe of shocks and providing a nearly constant rate of flow from the discharge.

25. Size of Delivery Air Chamber.—The proper size of an air chamber depends on the type of pump, the speed at which it works, the length of the discharge pipe, and the pressure head against which the pump works. For ordinary double-acting pumps working against moderate pressures and at ordinary speeds, the cubical contents of the air chamber should be not less than 3 times the piston displacement. For pressures of 100 pounds per square inch and upwards or for high piston speeds (as in the case of fire pumps), the capacity of the air chamber should be at least 6 times the volume of the piston displacement for a single stroke.

26. Loss of Air.—Under the increased pressure in the air chamber, the air is absorbed by the water and gradually passes off with it. In this way all the air will finally pass off and the chamber will be made useless if no means are provided for renewing the supply.

27. A simple device for maintaining the supply of air in the air chamber of large pumps is shown in Fig. 19. A piece of $2\frac{1}{2}$-inch wrought-iron pipe *c* about 30 inches long is connected to the end of the pump cylinder *a* in a vertical

position, by means of a gate valve *b*, or cock. A $2\frac{1}{2}$-inch **T** *d* at the upper end of this pipe is connected at one end of the run with a $1\frac{1}{4}$-inch check-valve *e* opening inwards, and at the other end with a $\frac{3}{4}$-inch check-valve *f* that opens outwards. The valve *f* is connected with the air chamber through the pipe *g*.

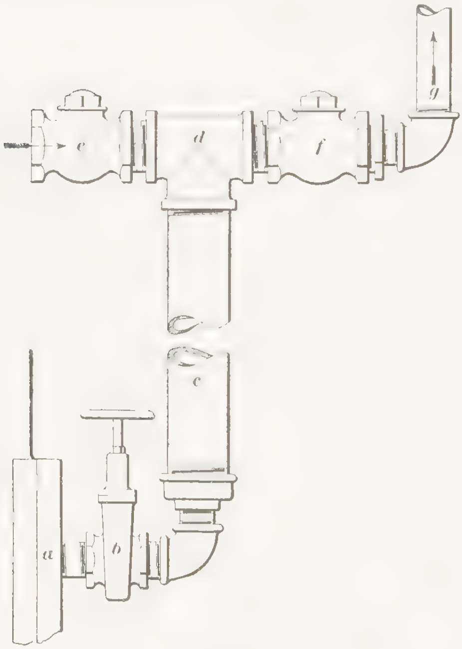


Fig. 19.

This air pump is operated as follows: When the pump is working, open the valve *b* to fill the pipe *c* with water; then partially close *b* until the check-valves *e* and *f* begin to work. This is easily determined by the click of the valves when seating. Its working may be described thus: When the valve *b* is opened, water fills the pipe *c* from the pump cylinder *a* during the discharge stroke of the pump. By partly closing *b* when *c* is full, the pump during the suction stroke will draw a part of the water from *c*, and air will flow in through *e* to take its place. During the next discharge stroke of the pump, more water is forced into *c*, driving the air out through *f* and *g* into the air chamber. If *b* is opened too wide, all the water will be drawn out of *c* during the suction stroke and air will be drawn into the pump cylinder from *c*; but by properly regulating the opening, a column of water is kept in *c*, which acts as a piston that moves with the strokes of the pump and pumps air into the air chamber.

28. Alleviator.—When pumps work under pressures greater than that due to a 350-foot lift, air chambers are not of very much service, owing to the fact that the air escapes from the air chambers either through the pores of the iron or at the joints, or it is absorbed and carried off by the water; in such a condition an air chamber gives the pump no relief whatever. To obviate this defect **alleviators** are used. An alleviator is shown in Fig. 20. It consists of a plunger *a* working through a water-packed stuffingbox. On top of the plunger are arranged springs that may be in the form of rubber buffers or helical coil springs. In the type shown rubber buffers *b*, *b* are used, which are confined by the tie-rods *c*, *c*, the yoke *d*, and the plates *e*, *e*. When the pressure in the pipe exceeds the working pressure, the plunger *a* is forced out through the stuffingbox and relieves the pump of the shocks that would otherwise occur. Alleviators may be placed anywhere on the delivery pipe, but are preferably placed in such a position that the direction of the moving water is in line with the plunger *a*.

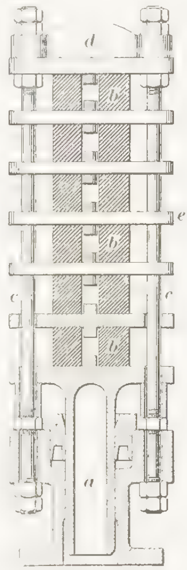


FIG. 20.

SUCTION AIR CHAMBERS.

29. Purpose.—With a long suction pipe or a pipe with numerous bends and valves, the resistance to the flow of the water through it will be considerable, and a great deal of force will be required to start and stop the water in it with each stroke of the pump. In some cases the force required is so great that the pressure of the atmosphere is not sufficient to set the column of water in motion quickly enough to fill the pump chamber as fast as the piston moves. This makes the action of the pump imperfect and causes a

H. S. V.—7

severe blow, called the **water hammer,** when the piston again meets the inflowing water.

30. The difficulty mentioned in Art. **29** can best be remedied by the use of a chamber, called a **vacuum chamber** or a **suction air chamber,** attached to the suction pipe as near the pump as possible. In its general form a vacuum chamber resembles an air chamber, but the pressure in it instead of being greater is always less than the atmospheric pressure. When the pump is drawing water, the air in the vacuum chamber expands and forces the water below it into the pump; at the same time the pressure of the atmosphere forces water in through the suction pipe to balance the reduced pressure in the vacuum chamber. The vacuum chamber is again partly filled and the air in it is compressed during the discharge stroke of the pump. It thus acts as a reservoir that receives from the suction pipe a nearly steady supply, which is given up intermittently to the pump.

31. Special Form of Suction Air Chamber.—Fig. 21 shows a special form of a suction air chamber in diagram-

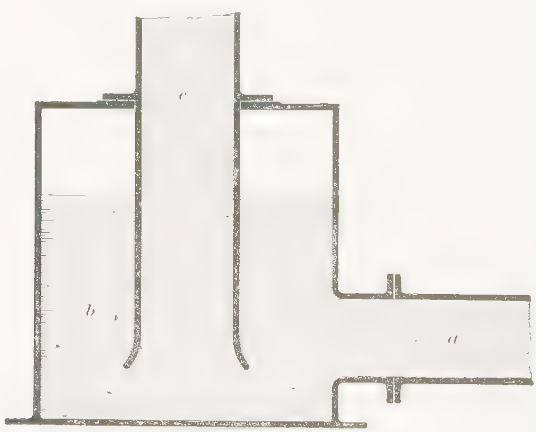


FIG. 21.

matic form. The suction pipe *a* connects to a suitable chamber *b*, which has a tube *c* projecting downwards to

within a short distance of the bottom. The tube *c*, which is called a **draft tube,** connects to the pump chamber. When water first flows into the chamber *b*, it entraps some of the air as soon as the water seals the bottom of the draft tube; this air is then compressed while the water flows up the draft tube, and by its expansion and compression permits a steady flow in the suction pipe.

32. Size of Vacuum Chambers.—For ordinary cases, the vacuum chamber may be made half the size of an air chamber working under the same conditions. A good rule is to make the cubic capacity of the vacuum chamber for a single pump twice that of the displacement of the piston for a single stroke.

33. Location.—Suction and delivery air chambers should, if possible, be placed at a bend in the pipe and close to the pump and in such a position as to be in line with the flow of water in the pipe. If placed at right angles to the flow of water, as in Fig. 18, their efficiency is somewhat impaired. Both suction and delivery air chambers should be provided with glass water gauges so that the height of the water can be determined at a glance. It is not customary to provide the air chambers of small pumps with water gauges.

PUMP FOUNDATIONS.

GENERAL CONSIDERATIONS.

34. The foundation for pumping machinery depends entirely on the type of pump. Generally speaking, much less foundation is required than for steam engines occupying about the same space. Direct-acting duplex pumps probably require the least foundation of any kind of steam pump, for here the piston and plunger motion is almost opposite and the balancing of the machine in line with the plunger motion is complete, and the strains due to reversing are contained almost wholly within the machine itself. Small duplex-pump foundations are made of a solid mass of brick or

concrete, while large pumps are often set on separate piers, one for the water ends and one for each pair of steam ends in case of a duplex, compound, or triple-expansion engine. Of course, the foundations must go down to sufficiently hard soil to bear up the weight of the pump, or if the soil be loose sand or gravel, the foundation must be spread out sufficiently to insure the pressure not exceeding, say, 1 ton per square foot. The foundation should go deep enough to allow the surrounding soil sufficient hold upon it to keep it firm and steady. The minimum depth for a small pump should not be less than 2 feet. Single-cylinder pumps require a somewhat heavier foundation than duplex pumps, owing to the greater shocks to which they are subjected.

35. Crank-and-flywheel pumps require considerably more foundation than direct-acting machines, on account of the much higher speeds possible and the weight and lack of balance of the reciprocating parts. Crank-and-flywheel pumps of the controlled-valve type, as the Riedler pumps, which usually run at a high speed, require foundations fully as heavy as those for steam engines of equal size.

MATERIAL AND FOUNDATION BOLTS.

36. Foundations should be built of hard brick laid in cement mortar, concrete, or, in the case of large pumps, of stone, if it can be readily secured. All pumps should be held down by foundation bolts. In the case of small pumps the bolts are provided with a steel or wrought-iron plate washer built solidly into the foundation, while large pumps have tunnels or pockets for access to the lower foundation washer and nut. If the foundation bolts are built in solid, box washers should be used.

FOUNDATIONS FOR LARGE PUMPS.

37. In the case of large vertical pumping engines, the masonry required to form the pump pit and to support the superstructure is of ample mass for all foundation purposes; in fact, large arched chambers and tunnels are often used to

save foundation materials in this class of pumping engine. These large pumping engines are often located at or near a water supply where the soil has not sufficient rigidity to support the weight. In this case piling must be resorted to, on which the foundation proper is constructed.

USE OF FOUNDATION TEMPLET.

38. A foundation templet should always be used in which the foundation-bolt holes are carefully laid off, preferably from the actual castings, and the various heights of bosses or thicknesses of casting through which the bolts pass are marked. The templet should be carefully set with reference to the suction and delivery connections, so that when the pump is set up, the fittings and pipes will connect up properly. In large pumps it is customary to arrange the pipe connections in such a way that a short space is left between the piping and the pump. This space is then measured after the pump and piping are in place, and a distance piece is made to suit the measurement and then put in place.

FOUNDATIONS FOR SMALL PUMPS.

39. Small pumps of the single-cylinder and duplex type are usually provided with two points of support only, one of which is rigidly bolted to the foundation, while the other is left free. This prevents the pump being thrown out of line, if properly constructed originally. When both the steam and water ends are bolted down, care must be taken not to twist or throw the pump out of line. In making the steam and water connections, the pipes should come fair to their connections and should not be sprung into place. Stresses on the pump structure due to winding foundation surfaces and sprung pipe connections should be guarded against, particularly with steam-thrown valves, as these are very sensitive and must be perfectly free. Any slight springing of the valve chamber will bind the valve and prevent its operating.

PIPING.

SUCTION PIPING.

40. Location of Pump in Respect to Supply.—Before a pump can be properly located, the location of the source of supply of the liquid to be pumped must be taken into consideration. Since the atmospheric pressure of 14.7 pounds to the square inch will balance a column of water 34 feet high, it is evident that with that atmospheric pressure the pump must not be placed more than 34 feet vertically above the surface of the water to be pumped. But since a perfect vacuum cannot be obtained by mechanical means, and since the flow of the water is retarded by friction in the pipes and passages, the limit of vertical lift by atmospheric pressure is reduced to about 28 feet at sea level in actual practice. The actual lift, precisely as the theoretical lift, varies with the atmospheric pressure, and hence will become smaller with an increase of altitude above sea level, since the air becomes lighter and its pressure less.

41. Run of Suction Pipe. The pump should be placed as near the source of water to be pumped as is possible, both vertically and horizontally. The suction pipe should be as straight as possible; if bends are necessary, they should be made by bending the pipe to a long radius or by using long-turn fittings. The suction pipe should be one diameter from end to end; all enlargements or reductions in size tend to disturb the uniform flow of the water so essential to a proper filling of the pump chamber. If from necessity the suction pipe is very long, it will be well to increase the size somewhat; the reduction at the pump chamber should then be made by a long conical fitting. For ordinary service pumps the diameter of the suction pipe should be such that the velocity does not exceed 200 feet per minute, assuming that the flow of water is constant. If the vertical lift be high, a suction air chamber should be provided; this will

add much to the uniformity of the pump supply. A foot-valve should also be provided when the lift is high.

42. Foot-Valves.—A foot-valve is a check-valve placed at the lower end of the suction pipe below the water level in the source of supply and opening towards the pump. Its purpose is to prevent the suction pipe emptying while the pump is at rest and to prevent the water in the suction pipe slipping back while running. When the water flows to the pump by gravity, a foot-valve is superfluous; but when the water is lifted by suction it is often fitted, since it will insure a prompt starting of the pump, providing that it is tight enough to hold the water in the suction pipe. In very cold weather and in exposed locations, the foot-valve constitutes an element of danger when the pump is out of use, since it prevents the emptying of the suction pipe. The water in the latter may freeze and burst the pipe. To prevent this, a drain may advantageously be fitted to the lower end of the suction pipe, which is used in cold weather to empty the pipe if the pump is to stand idle for a long time.

43. When foot-valves are used, a relief valve may advantageously be placed on the suction pipe. Generally, the suction pipe is made considerably lighter than other parts of the pump, and if the suction valves should leak when the pump is standing or if the priming pipe be left open, the full pressure of the delivery water will come on the suction pipe and foot-valve, which are not usually designed to withstand such pressures. The relief valve, which should be set to relieve the pipe at a pressure well within its safe strength, prevents overstraining of the suction pipe from this cause. Foot-valves should be chosen with the greatest care; they should be simple and, preferably, of the weighted-lift type or clack valve, and should have at least 50 per cent. excess of area over the suction pipe.

44. Settling Chamber.—If the water to be pumped is gritty or contains foreign substances, a settling chamber is sometimes used, especially when pumping water holding but

a small quantity of sand in suspension. This consists of an iron box conveniently arranged in a horizontal pipe. It is usually of large relative capacity, a settling chamber for a 2-inch pipe being 2 feet × 2 feet × 3 feet long. The pipes enter and leave from opposite sides and near the top. The increased volume of the large box allows the water to move very slowly across the box, giving the suspended sand time to settle to the bottom. The settling chamber should have a removable cover for the purpose of removing the settlings. This device is used on small pumps working on artesian wells.

45. Suction Basket and Strainer.—More universal arrangements for keeping back foreign matter from the working barrel of the pump are the **suction basket** and the **strainer.** The suction basket is usually placed on the bottom of the suction pipe and consists of a box variously shaped and perforated with strainer holes or provided with screens. The suction basket so placed is being replaced by a different form of strainer, which consists of a chamber placed in the suction pipe, located in an accessible position and provided with strainer plates so made that they can be readily removed for cleaning. This strainer is sometimes connected directly to the pump, but it should not be so placed that it will interfere with removing the water-cylinder heads. A short piece of pipe between the strainer and pump nozzle will avoid this interference. The objection to the suction basket on the bottom of the suction pipe is its inaccessibility for cleaning and inspection, a feature that is overcome by the strainer.

DELIVERY PIPING.

46. Run and Valves.—While the suction pipe is very important and must be most carefully laid out and has much to do with the location of the pump, the delivery pipe should not be neglected. A careful adjustment between the supply and delivery pipes should be made in order to produce the

best effect of the whole plant. The delivery pipe should as far as possible be a plain, straight pipe from pump to terminal; when bends are necessary, they should be by as long sweeps as possible. A gate valve or check-valve should be placed near the pump. The check-valve serves the double purpose of relieving the pump of pressure when starting up, allowing it to take hold of the water more quickly, and also holds the water back from the pump when inspection and repairs to the water end are necessary. If a check-valve is not used, a gate valve should be placed at or near the pump delivery to hold back the water in case of repairs to the pump end or accident. This valve should always be a straightway gate valve giving the full clear opening of the pipe.

47. Velocity of Flow.—The velocity of the water flowing through the delivery pipe for direct-acting pumps should not much exceed 330 feet per minute, while for large crank-and-flywheel pumping engines the velocity of water in both suction and delivery pipes is about 300 feet per minute. If the suction pipe is made small, the pump will fail to fill and the plunger will strike the incoming water on its return stroke, producing a violent and dangerous shock. If the delivery pipe is made small, the cost of power required to force the water through the pipes at a high velocity will very quickly overrun the interest and depreciation on a larger pipe.

AUXILIARY PIPING.

BY-PASSES.

48. Water-End By-Pass.—**By-pass pipes** are pipe connections from above to below the delivery-valve deck and are of much more use on crank-and-flywheel pumps than on direct-acting machines. In the case of compound pumps, when starting up, the force of the full steam pressure on the high-pressure piston is not sufficient to move the plungers

against the resistance due to the head of the water in the delivery pipe; but by opening the valve (which, by the way, should always be a gate valve) in the by-pass piping, the pressure on the plungers is relieved for a sufficient number of strokes to allow the steam to reach the low-pressure piston, when the combined force of the two pistons will do the work and the by-pass pipe can be closed.

49. By-pass water pipes have another function on crank-and-flywheel pumps. Unless these machines are fitted with very large flywheels, their limit to slow running is often not as low as desired. By opening the valve in the by-pass pipe, part of the water can be returned to the pump chamber and the amount of water actually pumped reduced to any desired quantity permitted by the size of the by-pass. It should not be overlooked that this is accomplished at a very considerable loss of efficiency, because it takes the same power to move the by-pass water as it does to do the actual pumping, comparing equal quantities. By-pass pipes are usually made $2\frac{1}{2}$ per cent. of the plunger area.

50. Steam-End By-Pass.—It is common practice to fit the steam cylinders with by-pass pipes, allowing high-pressure steam to act on the low-pressure piston in starting, but these pipes are usually so small, compared with the diameter of the low-pressure piston, that the by-pass is unable to hold any pressure behind the low-pressure piston when it is moving. By-pass steam pipes have their proper use in warming up the low-pressure cylinder and connections, and in the case of crank-and-flywheel pumps to move the high-pressure crank off the dead center.

PRIMING PIPE.

51. The **priming** or **charging pipe** is a small pipe run from the delivery pipe beyond the check-valve or delivery gate valve to the suction chamber of the pump. It is particularly useful in the case of long suction lifts to fill the working chamber and suction pipe with water, taking up all

clearances and helping the pump to take hold of the water quickly. This pipe may be from $\frac{3}{4}$ of 1 per cent. to 1 per cent. of the area of the plunger; its size is a matter of little importance, but it should be large enough to fill the suction pipe and pump chamber in a reasonable time, which will depend somewhat on the size and design of the pump chamber and the length of suction pipe. A pipe much larger than 1 per cent. of the plunger area will be required in the case of long inclined or horizontal suction pipes.

WASTE DELIVERY PIPE.

52. A **waste delivery** or **starting pipe** that can be led into any convenient place of overflow should be provided so that the pump, in starting, can free itself of air, for it often happens that a pump refuses to lift while the full pressure against which it is expected to work is resting on the delivery valves, for the reason that the air within the pump chamber is not dislodged but only compressed and expanded again by the motion of the plunger. A pump in this condition is said to be **air bound.** It is well in this case to run with the delivery pipe empty until the air is expelled and the water flows into the suction end of the pump. The waste delivery pipe is fitted with a valve and connected to the delivery pipe close to the pump. When the water flows to the pump and is discharged into the delivery pipe, the valve in the waste delivery pipe is to be closed.

AIR DISCHARGE VALVES.

53. When a check-valve is not used in the delivery pipe and the space between the suction and delivery valves is large and the delivery pipe is full of water, the pump will often refuse to start the water in the suction end, owing to compressed air being trapped between the water in the delivery pipe and the delivery valves. Air discharge valves performing similar service to the waste delivery or starting

pipes may then be used to allow the compressed air to escape and a vacuum to be created when the plunger is withdrawn from the pump chamber. In small pumps of the direct-acting type, a petcock is usually fitted for this purpose to the cover directly above the delivery valves.

GENERAL PIPING ARRANGEMENT.

54. Fig. 22 shows a good arrangement of a pump in relation to the water supply and of the pipe connections. The

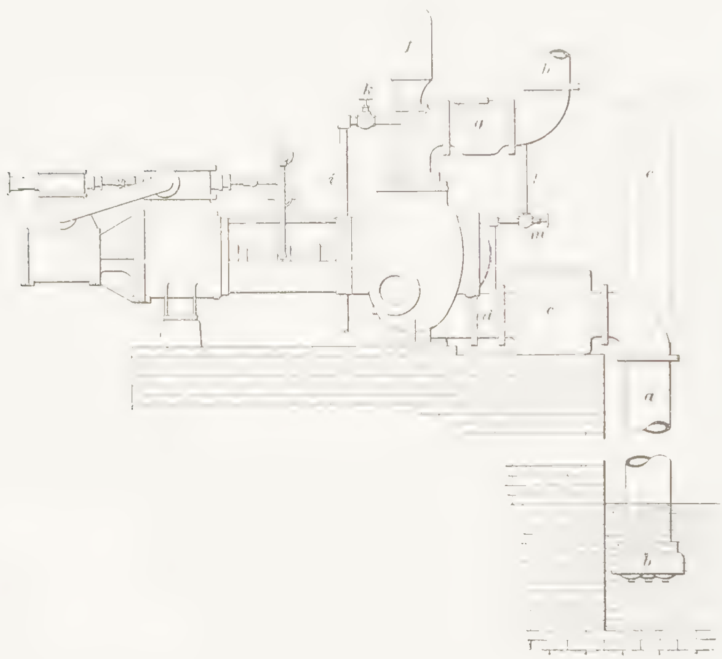


FIG. 22.

suction pipe *a* is fitted with the foot-valve *b* and has a strainer *c* placed close to the pump, from which it is separated by the short distance piece *d*. When the vertical lift is short, say not over 10 feet, and the pump is placed close to the source of supply, a suction air chamber is seldom necessary, but when the lift exceeds 10 feet or when the pump is

at some distance from the water supply, a suction air chamber becomes a necessity. With a vertical suction pipe as shown, the suction air chamber may be made as shown by the dotted lines at *c*. An air chamber *f* is placed on the delivery between the delivery check-valve *g* and the delivery valves. The waste delivery or starting pipe *i* is connected to the delivery between the delivery valves and the delivery check-valve *g*. It is fitted with the valve *k*. The delivery pipe *h* is connected to the suction pipe close to the pump, in this case to the distance piece *d*, by the priming pipe *l*, which is fifted with the stop-valve *m*.

PROVISION FOR DRAINAGE.

55. Proper drain.pipes and drain valves should be provided for all parts of the pump, the pipe connections, strainers, etc., in short, for all parts in which water may remain when the pump is not in use and will give trouble by freezing.

Provision for draining the suction valve deck and delivery valve deck is sometimes made by drilling a small hole through the decks; this practice, while simple and cheap, leads to a loss in efficiency, however, since some of the water is constantly flowing back into the suction chamber.

PUMP MANAGEMENT.

INTRODUCTION.

56. If a pump has been properly selected for the service and has been properly designed, built, and erected, it should perform its work without any trouble. All pumps when new are stiff and cranky in their actions, particularly direct-acting pumps. They should be run slowly for a considerable time, and many defects in their action which at first gives rise to alarm will then gradually disappear. Crank-and-flywheel pumps act more smoothly from the start, but do

not come to a proper bearing more quickly or quite as quickly as the direct-acting pump. Crank-and-flywheel pumps usually require considerable skill and study to reduce them to successful working order, as conditions arise that further disturb the lack of harmony between the flywheel and water, and it often taxes the skill of the experienced engineer to make an amicable adjustment between the two opposing forces.

57. Having reduced the pump to satisfactory operation, the attention of the operator should be directed to its maintenance at the least possible expenditure. Each item of expenditure should be separated from the whole and studied independently for the purpose of reducing it to a minimum consistent with the proper maintenance of the plant. The expenditure should at all times be regarded as the item by which interest or dividends are being earned and should not be allowed to become greater.

58. Losses in efficiency arise from wear, from loss of proper adjustments, and from the wrong timing of the various movements that control the distribution of steam, by leakages, by decreased mechanical efficiency due to lack of alinement, by accumulations of foreign matter on and in condenser tubes, suction strainer, and foot-valves, suction and delivery pipes, and in many minor directions. In many plants it is of the utmost importance that they should not be interrupted; it is then the duty of the engineer to predict all possible events that might cause an interruption and have a well-planned line of action prepared so that he may act quickly and with decision to the end of keeping his plant always at work and at the highest efficiency. This plan of action will entail considerable work, study, and, perhaps, some expense in preparation to meet possible contingencies that may never happen; nevertheless, it is well to be ready for any emergency when handling steam machinery and particularly steam pumping engines.

59. In the management of pumps it must be considered that nearly every installation has its peculiarities, some of

which are sometimes not discovered until after the machine is put in service and then perhaps require expensive additions and alterations to meet them. An exhaustive study of existing conditions and resultant conditions when the pump starts to work cannot be too strongly urged.

STARTING PUMPS.

IMPOSSIBILITY OF SPECIFIC RULES.

60. Pumps differ so much in their construction and design that it is entirely impossible to lay down specific rules that will be applicable to every pump. For this reason only *general* rules are here given, which must be modified by the pump attendant to suit every specific case.

GETTING A PUMP READY.

61. Getting Up Steam.—Considering a new steam pump, after it has been properly erected on a suitable foundation and all the pipe connections have been made, the first step in starting the pump is to get up steam in the boiler or boilers in the same manner as is done with boilers supplying steam for any other purpose.

62. Since the boilers are generally in charge of the same person that attends the pump, the general treatment of the pump and the boilers, while steam is being raised, will be considered together. After the steam piping is in place, but before it is finally connected to the pump, all valves in it should be opened wide; while steam is being raised the pistons and valves should be removed from the steam end of the pump so that there is a clear passage for the steam from the boiler to the exhaust after the steam pipe has been connected to the pump.

63. Blowing Out the Steam Piping.—The fires should be started very slowly under the boiler; all the binding

bolts throughout the boiler setting should be perfectly loose and free. If this precaution is neglected, buckstaves or cast-iron fronts will be broken by the expansion of the setting. The guy rods on iron stacks should also be slacked off; in fact, every part that will expand when the plant is started up should be liberated. Before the steaming stage is reached, large volumes of heated air will be driven through the pipes, warming them up gradually. When steam begins to rise, it should be allowed to blow through the piping and valves quite liberally, the object being to clear the piping of sand, grit, and all other foreign matter collected therein during erection. The piping having been blown out thoroughly, steam is shut off and the piping is then connected to the pump.

64. Blowing Out the Cylinder. When the pressure in the boiler has been raised to the working pressure, the cylinder heads should be put on, still leaving the pistons and valves out of the cylinders. The stuffingboxes should be closed, which is most conveniently done by placing a piece of board between the stuffingbox and the reversed gland and then setting up the nut on the stuffingbox studs. When the gland is drawn home by a nut outside of it, a circular piece of pine board may be placed between the end of the gland and the inside of the nut in order to close the opening through which the piston rod passes. The steam may now be turned on the main steam pipe leading to the pump; by opening the throttle valve wide at short intervals, the sand and scale in the ports and other passages and spaces of the steam end can be blown out. After the cylinders have been blown out, the heads and covers should be removed, and all foreign matter blown into the corners and chambers of the cylinders should be removed by hand. The pistons, valves, cylinder heads, and other covers can now be put in place.

65. The blowing out of the pipes and cylinders after erection is often neglected or but imperfectly done, with

serious consequences to the machine; it cannot be too thoroughly done, particularly in that type of pump where the steam ports and exhaust ports are on top, for in this particular case the sand and grit are deposited in the bottom of the cylinder for the piston to ride upon. If more attention were paid to the thorough cleaning of all steam spaces, we would hear less of cylinders and pistons being cut.

66. Keying Up.—If the pump is of the crank-and-flywheel type, it should be turned a complete revolution by hand to insure that everything clears properly and that no tools or materials used during construction or erection have been left within the machine. The adjustment of all journals, pins, and bearings should then be made. With gib and key ends, it is usual to drive down the key with a soft hammer (lead hammer) until it is home, mark it, drive it back, and then tap it down to within $\frac{1}{8}$ inch of the mark. With wedge ends the wedges usually have an inclination of $1\frac{1}{2}$ inches per foot and the adjusting screw 8 threads per inch. The wedge is drawn up solid and then the adjusting screw is turned back about 20° and locked. Bolted connecting-rod ends are allowed about $\frac{1}{200}$ inch play, using liners and setting the bolts up solid. Main bearings can be adjusted best when the machine is in motion.

67. Packing Rods and Stems. — The packing of all rods and stems is the next step. If fibrous packing is used, the boxes should be filled full and the glands tightened down very moderately. The tightening of the glands can best be done when steam is on and the machine is in motion, when they should be tightened only sufficient to stop leakage and no more. When excessive tightening is required to stop leakage, the packing should be completely renewed. Some pumps are fitted with metallic packings. These packings are usually fitted up by specialists who fully guarantee them, and their directions for use should be carefully followed; in case of failure or unsatisfactory results, the makers should be consulted.

68. Oiling.—The oiling of the machinery is the next step and is a very important one. All rubbing surfaces should be provided with suitable oiling devices appropriate to the particular place and service. The quality of oil should be carefully selected to suit the velocity and pressure of the rubbing surfaces on which it is used. For use within the steam cylinder, heavy mineral oil is the only oil capable of withstanding the high temperature, and in starting up new pumps only, the best quality should be used, regardless of price. A liberal use of this oil for the first month will go far towards reducing subsequent oil bills.

69. The pumping engine, unlike many other types of engines, must often run continuously and without interruption for a month or even longer at a run. This requires that all oiling devices be so arranged that they can be supplied and adjusted while the machine is in motion. It is a good plan to provide two separate sets of oiling systems for all the principal journals, the idea being that if one fails the other can be used while the disabled one is being overhauled. All oil holes should have been filled with wooden plugs, bits of waste twisted in the hole, or some other protection, while the machine was being erected. These should now all be removed and all oil holes and oil channel thoroughly cleaned out. Bearings should be flooded with oil at first to wash out any dust or grit that may have reached the rubbing surfaces.

70. Having turned the machine by hand and inspected all locknuts, setscrews, and clamp screws, the engine may be put under steam. If provided with hand starting gear, this should be used for a sufficient number of turns to make sure that the machine is free from water that may have accumulated in the pipes or clearance spaces. All drain cocks should be wide open when starting and relief valves should be adjusted to blow at the proper pressure. If the engine is condensing, connections from the exhaust port to the condenser should be made absolutely tight. If an independent condenser is used, it should be started before

the main pump is started and a vacuum obtained in advance.

71. So far only the steam end of a large crank-and-flywheel pump has been considered. With the direct-acting single or duplex steam pump, the same general method of procedure should be followed. It may be mentioned here, incidentally, that the direct-acting pump is not so liable to an accident in starting as the crank-and-flywheel pump on account of the absence of kinetic energy stored up in a moving flywheel. This energy when given out by reason of an obstruction in the water end that prevents the free passage of water will greatly increase the pressure, especially when the obstruction occurs near the dead-center positions of the crank. The increased pressure thus produced may easily run up high enough to burst the water end.

72. Using the Dash Relief Valves.—In starting a direct-acting pump when dash relief valves are fitted, they should be closed in order to keep the pistons as far from the heads as possible, for in new installations the unexpected is likely to happen at the water end, and to prevent danger of a breakdown, as in case of a sudden lunge of the pistons, all the margin possible to keep them from striking the heads should be gained.

73. Condition of Water End When Starting.—Assuming that the plungers and plunger rods are packed and the plunger grease cups filled, the water end should be ready to start; if the machine is compound or triple-expansion, the water by-pass valves must be opened until the machine has made a sufficient number of strokes to bring the intermediate and low-pressure cylinders into action, when the by-pass valves should be closed. The suction pipe from the foot-valves to the delivery valve deck must be absolutely tight; anything short of this will cause the water end to refuse to work satisfactorily. All the suction valves and delivery valves should seat fairly and tightly. Care must be taken that there is no obstruction in the delivery pipe, such as a

closed valve, as pumps usually have sufficient margin in the driving force over the resistance to burst the water end, particularly if the momentum of a flywheel be added to it.

74. Pressure gauges should always be attached to the suction and delivery pipes, and they should be carefully watched during the process of starting, as trouble at the water end will be promptly recorded by the gauges. The lower end of the suction pipe should be kept well under water, as a slug of air taken into the pump may cause a violent jumping and in a direct-acting pump possibly a striking of the steam pistons against the heads.

75. Watching the Air Chamber.—The delivery air chamber should be carefully watched during the starting and running. This should be provided with a gauge glass showing the height of the water and extent of the pulsation. The air chamber should be charged with air when the air in the chamber is lost, as shown by the rise of the water in the gauge glass. Large pumps are usually supplied with an air charging pump that is attached to and driven by the main pump, or an arrangement of pipes and valves is sometimes improvised for this purpose. In very large pumping plants, an independent air compressor or locomotive air pump is often used for this service. A very good idea of the internal working of the pump can be obtained by placing the ear against the pump chambers; the seating of the valves can then be distinctly heard, and if there is any leak past either the suction or the delivery valves, it, too, is quite audible.

DEFECTS IN PUMPS.

SUCTION-END TROUBLES.

76. The most common causes of pump failures are leaks below the suction valves. These may be at the joints or along the suction pipe or in the pump chamber, and may be due to imperfect connections, leaky chaplets, shifted cores, blowholes, corrosion, or cracks from frost.

77. Small leaks in the suction end which are not sufficient to cause entire failure will cause the piston to jump, i. e., move suddenly, during the first part of the stroke. Leaky valves and plungers reduce the capacity of the pump; if this is the case, they should immediately be refitted and repacked. It is always best to have hot water flow to the pump by gravity; if it is necessary to lift it and the pump works with a jerky action, the lift is too high for the temperature, and one or the other must be reduced. In pumping from wells, care should be taken that the pump is near enough to the water to prevent the water falling below the maximum lift by suction.

78. If the pump pounds soon after the beginning of a stroke, when running fast, it shows that the pump chamber is not filling and that the plunger is striking the incoming water on its return stroke. A suction air chamber will help to remedy the evil. Obstructions under the suction or delivery valves will cause a very decreased output or total failure. A suction strainer or end of suction pipe becoming embedded in sand or clogged with foreign matter will cut off the supply from a pump.

79. Air pockets under the delivery valve deck, caused either by bad design or a shifting core, will very much reduce the capacity and efficiency of a pump. The effect of the air pocket is to entrap air, which is compressed to delivery-water pressure and expands again on the suction stroke. If the relative capacity of the pocket to the plunger displacement is sufficient, the entrapped air will expand to atmospheric pressure, reducing the suction lift to zero; this defect, however small, will always reduce the suction lift and is not easy to remedy; its existence should always be cause for the rejection of a pump.

80. Pounding in pumps is sometimes caused by the water lagging behind the plunger, due to the friction of a small, long, horizontal suction pipe. When suction pipes have a long horizontal run, they should be one or two sizes larger.

DELIVERY-END TROUBLES.

81. Pumps sometimes fail when the full head is resting upon the delivery valves by the air between the suction and delivery valves being expanded and compressed by the motion of the plunger. Air cocks should be provided close up under the delivery decks for discharging the air until the plungers have caught the water. If only a simple cock is fitted, it must be opened during the delivery stroke only and closed shortly before the suction stroke commences. This is to be repeated until a steady stream of water issues from it during the delivery stroke. An automatic air valve, which is simply a small spring loaded valve opening outwardly and closing automatically during the suction stroke, is preferable; this valve should be secured to its seat after a steady stream of water issues during the delivery stroke. Violent jarring and trembling of the pump arises from the delivery air chamber becoming filled with water. It should be recharged with air by means of the air-charging pump, a near-by air compressor, or by a hand air pump.

STEAM-END TROUBLES.

82. The steam end of pumps should not be taken apart needlessly, especially the steam end of direct-acting pumps with steam-thrown valves, as their action is quite complicated, and a very slight misadjustment will cause a failure. If at any time it becomes necessary to dismantle the pump, all the parts, if not already marked, should be plainly marked with steel letters or numbers, rather than with a prick punch or chisel, and suitable gauges, by which all parts can be returned to their correct relative positions, should be made, if this is deemed advisable. In many duplex pumps there are very slight differences in the two sides; for instance, the crossheads that drive the valve levers are not keyed in exactly the same position on the piston rods and the rods are not interchangeable; the pump will not run successfully if they are interchanged. In some pumps with steam-thrown valves, the valve chests are bolted to the cylinders,

and are reversible so far as fitting and bolting goes, but the auxiliary ports are not reversible and will be shut off in both valve chest and cylinder by reversing the chest. In placing the gasket between the valve chest and cylinder of pumps with steam-thrown valves, care should be taken to cut passages through the gasket for the auxiliary ports. The valve levers, pins, and all connections between the piston rod of one side of a duplex pump and the valve of the opposite side should be kept in good condition, as the failure of these parts will cause a serious accident.

83. On duplex pumps the amount of lost motion between the valve stem and the valve should be very carefully adjusted; too little lost motion will cause short stroking, while too much will allow the pistons to strike the heads. If the pistons strike the cylinder heads, the dash relief valves, if fitted, should be closed until the stroke is shortened sufficiently for the pistons to clear the heads. If the stroke becomes too short, the opposite course should be followed. If no dash relief valves are fitted, the lost motion should be made smaller in case the pistons strike the heads.

84. When a compound pump is fitted with a cross exhaust and it is seen that the pump is unable to complete its full stroke, the valve in the cross exhaust should be opened.

TESTING PUMPS FOR LEAKAGE.

85. Testing the Suction Pipe.—Leaks are the most troublesome and most frequent sources of loss of efficiency in pumping machinery. Leaks in the suction pipe or suction system affect the pump most and often cause its complete failure. These leaks can sometimes be detected by the ear, or the flame from a common tallow candle will often locate a leak in the suction by being drawn towards the hole by the air. Sometimes these leaks are very numerous, but so small that any one of them would be difficult to locate and be of small importance; at the same time, their combined effect may be sufficient to seriously affect the working of the

pump. The best way to locate these leaks, which may be at the joints or along the body of the pipe, is to stop up the inlet end of the pipe, uncover it completely, and then put a water pressure on it, say from 40 to 50 pounds per square inch. Any leaks, however small, will then be readily detected. The suction pipe should always be tested for leaks before it is covered, if laid in a trench or otherwise made inaccessible, because it must be made tight before the pump will work successfully.

86. Delivery Pipe Leaks.—Leaks in the delivery pipe, while common and at times more difficult to remedy than leaks in the suction, are plainly evident. They do not affect the action of the pump or its efficiency to any extent, the loss being exactly proportional to the magnitude of the leak.

87. Repairing Leaky Pipes.—Probably the most satisfactory method of procedure in case a leaky section of pipe is discovered is to discard it and replace it with a new one. Circumstances, however, do not always permit this to be done, and then temporary repairs should be made. The manner of making the repair obviously depends on the position and extent of the leak and calls for the exercise of judgment and some skill.

88. Small leaks in the form of pinholes in the suction pipe can generally be stopped effectually by a thick coat of red-lead putty spread over the pipe where the leaks occur. This should be covered with several layers of canvas covered on both sides with red-lead putty and wound as tightly as possible around the pipe. The canvas should then be secured by wrapping it with strong twine or annealed copper wire, put on as tightly as possible. If the suction pipe is split, it is usually well to cover the split part with a piece of sheet metal, preferably sheet lead, bent to the curvature of the pipe and put on with red-lead putty. The canvas should be wrapped over this.

A permanent repair in case of pinholes can be made by drilling out the pinhole with a twist drill and tapping out

the hole. A closely fitting threaded plug of soft steel or wrought iron is then screwed in and the end riveted over.

89. Small pinholes in delivery pipes can often be stopped up by the same means given in Art. **88** for suction pipes. If the leak is extensive, however, it will generally be necessary to use a **pipe clamp.** Such clamps may be made in a good many different ways, according to the location and extent of the leak and the facilities for repair. One of the simplest pipe clamps is shown in Fig. 23. It consists simply of a piece of sheet iron or sheet steel of sufficient width to cover the leak and bent to the form shown. A piece of sheet packing, which may be covered with red-lead putty to advantage, is placed over the leak and the pipe clamp is then placed over this and the ends drawn together by the bolt shown.

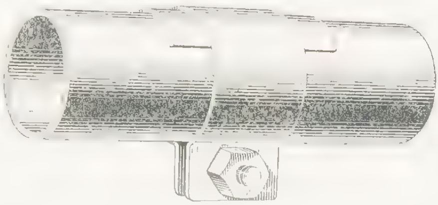


FIG. 23

The clamp shown in Fig. 23 is only adapted for small pipes. For large pipes the clamp must be made in two halves.

90. Testing Air Chambers.—Air chambers must be absolutely tight. They are usually tested by closing all openings and then pumping air into them until the working pressure is reached, as shown by a pressure gauge. After 24 hours this gauge should show no reduction of pressure. If the air chamber does not pass this test, the leaks may be discovered by filling it with water subjected to the working pressure. If there are a number of leaks, the chamber should be condemned; if only a few small leaks exist, they can usually be effectually stopped by drilling a hole at the leak and screwing in a plug.

91. Leakage of Pistons and Plungers.—The plungers of inside-packed or center-packed plunger pumps should be

tight themselves, besides making a tight joint through the stuffingboxes, in order that water may not pass from one side to the other. The manner of testing will depend on their design, the general method of procedure being the subjecting of one side of the plunger to an air pressure or hydrostatic pressure at least equal to the working pressure. If leaks are discovered, judgment has to be used as to the manner of repairing them or whether to condemn the plunger. In some designs of inside-packed and center-packed pumps with closed hollow plungers, the weight of the plunger is so proportioned to its displacement as to relieve the stuffingboxes of nearly or quite all of its weight; it is then important that they be absolutely water-tight.

92. Leakage Past Pistons and Plungers.—With piston pumps and inside-packed plunger pumps there is liable to be considerable unnoticed leakage. If it is extensive, it can be heard by placing the ear against the pump chamber. It is best with this style of pump to make regular inspections for leakage past the plunger or piston, providing suitable pipes and apparatus by means of which pressure can be put on one side of the packing or piston while the other side is exposed for inspection. With outside-packed plungers there can be no unobserved leaks past the plungers, and this is the principal reason for their use.

93. Leaks Past the Valves.—Leaks past the suction and delivery valves can readily be tested when the piston or plunger is being tested for leaks past them. The delivery and suction valves should be tested separately; the fact that the column of water in the delivery pipe does not drain out while standing is not proof that both sets of valves are tight, since either set will support the water while the other set may be leaking badly.

94. To test the suction valves for leakage, disconnect the suction pipe or take any other convenient steps that will allow the leakage to be seen. Fill the delivery pipe full of water, having removed enough delivery valves to allow the pressure to reach all the suction valves, and observe which

valves, if any, are leaking. When there is a valve in the delivery pipe, this may be shut and water pumped into the pump cylinder with a small force pump, running the pressure up to the working pressure. Care must be taken, by removing delivery valves if necessary, that the pressure reaches all the suction valves.

95. The delivery valves can be tested by filling the delivery pipe or by closing the valve in the delivery pipe and pumping water into the delivery pipe between its valve and the pump delivery valve. The pump chamber must be open so that the leaks can be seen.

SURGING OF WATER IN PIPES.

96. By surging of the water flowing through pipes is meant that its velocity of flow not only is not constant, but that the direction of flow reverses for a short period. This condition often exists in pumping machinery having very long suction or delivery pipes. It may occur either in the suction pipes or in the delivery pipes, being, however, most severe in the latter. Crank-and-flywheel pumps, owing to the variation in the piston speed between the beginning and end of the stroke, are particularly liable to cause surging, which is due entirely to an irregular delivery.

97. Duplex direct-acting pumps, owing to the uniformity of delivery and the absence of heavy weights, such as flywheels, are little liable to cause surging, and when liquids must be moved through long mains, an instance of which are the long oil pipe lines, this pump is chosen. Crank-and-flywheel pumps forcing water through very high delivery pipes, as occurs in mine work, are seriously affected by the surging of the water. Air chambers do not help matters, but probably aggravate them by forming an elastic cushion for the column of water to rebound from. The effect of surging water is to vary the pressure on the pump and mains, sometimes from zero to twice the pressure due to the vertical height, resulting in broken pump chambers,

pipes, and not infrequently in damage to the working parts of the pump, for the actual resistance to these shocks is not met until they arrive at the flywheel rim.

98. The remedying of surging is not easy of attainment. Air chambers placed along the delivery pipe at intervals are employed occasionally, the aim being to break up the vibrations of the surging water and get them out of step or out of harmony with the motion of the pump. Alleviators are sometimes used in place of air chambers to relieve the shock, and not being so elastic do not encourage surging to the extent that air chambers do. When for economical reasons it is desired to use the crank-and-flywheel pump, the variations in pressure and the liability to surging can be very much reduced by using the three-throw crank with the pins set at 120° from one another.

99. Surging in long suction pipes is liable to occur especially when the water flows to the pump by gravity; this is not so difficult to overcome or so serious in its effects as surging in the delivery pipe, for the reason that the direction of the force resulting from the surge is through the pump valves and into the delivery, or in the natural direction of the water, while the shock due to surging in the delivery pipe comes against the valves and must be withstood by the machinery.

100. To prevent shocks due to surging reaching the machinery, a liberal sized air chamber is needed on the suction main near the pump, and in addition spring-loaded relief valves may also be fitted to the main. These relief valves simply limit the pressure due to an unusually heavy surge that cannot be taken care of by the air chamber.

PUMPING A MIXTURE OF WATER AND AIR.

101. In mine and artesian-well work, large quantities of air are often mixed with the water, due to local disturbances in the source of supply, such as water discharging into it in the form of spray. When such a mixture of air and water

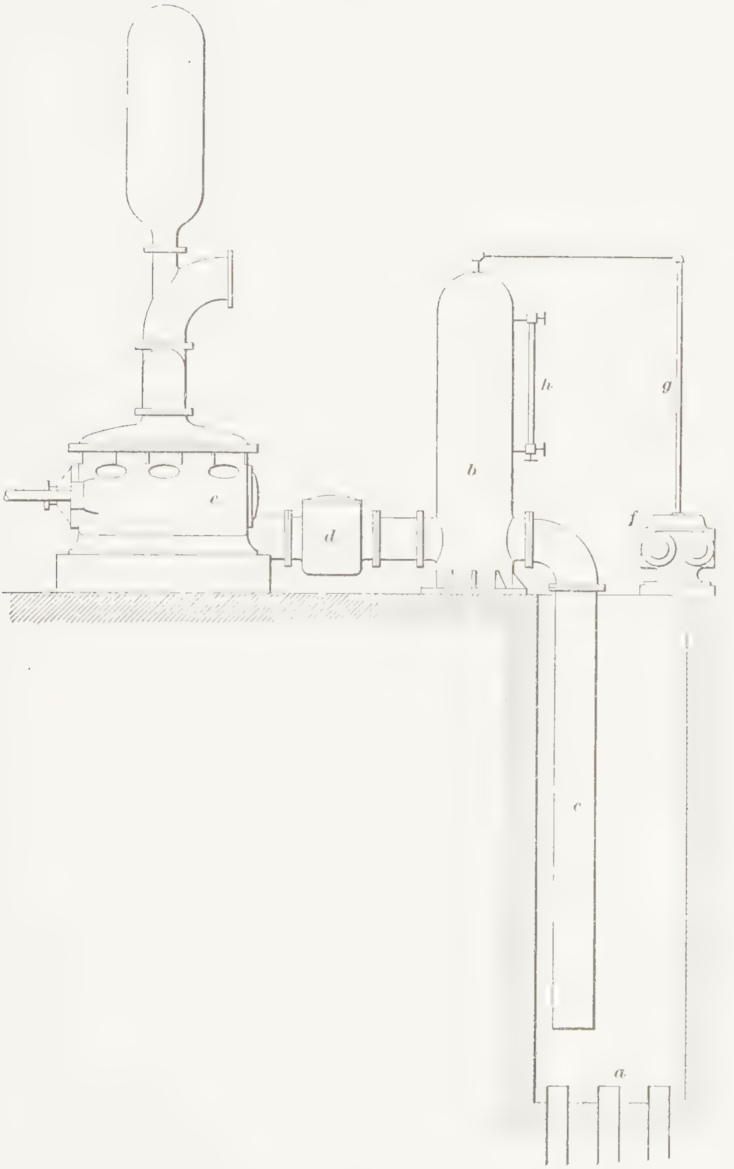


FIG. 24

is pumped, the pump will have a jerky motion, that is, instead of moving steadily it will move in jumps, and in the case of direct-acting pumps there is danger of striking the cylinder heads. Besides, on account of the uneven discharge there will be violent disturbances in the delivery pipe. The only effectual remedy is to remove the air before it arrives at the pump.

102. Fig. 24 shows the installation of a pump taking its water from an artesian well *a*, the water being highly charged with air and gas. A large suction air chamber *b* is put into the suction pipe *c*; the water passes through the strainer *d* to the pump *e*. A vacuum pump *f* is connected by the pipe *g* to the top of the air chamber and not only maintains a vacuum in the chamber, but draws the air and gas out of the water in the chamber and before it reaches the pump. The gauge glass *h* not only shows the height of water in the air chamber, but also allows the bubbles of air and gas rising through the water to be seen. The vacuum pump is simply an ordinary steam pump pumping air instead of water; it is running constantly and its speed is regulated to suit the height of the water in the air chamber.

SETTING THE VALVES OF DUPLEX STEAM PUMPS.

103. The steam valves of duplex pumps have no outside or inside lap, consequently when in their central position they just cover the steam ports leading to opposite ends of the cylinders. With all these valves a certain amount of lost motion is provided between the jam nuts and the valve. This lost motion in small pumps is within the steam chest, while in large pumps it is outside and may be adjusted while the pump is in motion. The first move in the process of setting the valves of duplex pumps is to remove the steam-chest bonnets and to place the pistons in their mid-stroke position. To do this, open the drip cocks and move each piston by prying on the crosshead, but never on the valve lever, until it comes into contact with the cylinder head.

Make a mark on the piston rod at the steam-end stuffingbox gland. Move each piston back until it strikes the opposite head, and then make a second mark on the piston rod. Half way between the first and second mark make a third one. Then, if each piston is again moved until the last mark coincides with the face of the gland, the pistons will be exactly at their mid-stroke position. After placing the pistons in their mid-position, set the valves central over the ports. Adjust the locknuts so as to allow about $\frac{3}{16}$ inch lost motion on each side. The best way of testing the equal division of the lost motion is to move each valve each way until it strikes the nut or nuts and see if the port openings are equal. When the port opening has been equalized, the valves are set. The valve motion need not be and should not be disturbed while setting the valves. Too much lost motion will tend to lengthen the stroke and may cause the piston to strike the cylinder heads, while on the other hand when there is not enough lost motion, the stroke will be perceptibly shortened. The proper amount of lost motion to give a certain length of stroke can only be found by trial for each particular pump.

104. If only one valve of a duplex pump is to be set, bear in mind that it is operated by the piston of the opposite pump. Place that piston in its mid-position and then set the valve as previously explained.

PUMPS.

(PART 3.)

CALCULATIONS RELATING TO PUMPS.

DISPLACEMENT.

1. The **displacement** of a pump for a single stroke is the volume of water that would be displaced (that is, driven out of the cylinder) by the piston or plunger during that stroke.

In calculating the displacement of a pump in a given time, care must be taken to consider the number of strokes during which water is discharged. Thus, for a single-acting pump, water is discharged only when the piston moves in one direction; and with the double-acting pump the number of strokes during which discharge occurs is equal to the total number of strokes that the piston makes. With a duplex double-acting pump, it is customary when giving the number of strokes per minute to refer only to the number of strokes made by one piston, which, obviously, is only one-half the total number of strokes made. As practice varies, however, among engineers in this respect, it is best to find out in each case, by inquiry, whether the number of strokes of one piston or of both pistons in a given time is meant when the number of strokes is given. In the case of a crank-driven pump, for a single single-acting pump the strokes will be equal to the revolutions of the crank; for a single

double-acting and a double single-acting crank-driven pump the strokes will equal twice the number of revolutions; for a triplex single-acting crank-driven pump the strokes will equal three times the number of revolutions; and for a triplex double-acting pump, six times the number of revolutions.

2. The displacement of a pump in a minute in cubic feet, gallons, or pounds is given by the following rule:

Rule 1.—*Multiply the length of stroke in inches by the mean effective area of the pump piston or plunger in square inches and the number of strokes per minute. The product is the displacement in cubic inches. To reduce the displacement to pounds, multiply by the weight of a cubic inch of the liquid pumped; to reduce to cubic feet, divide the displacement by 1,728; to reduce to Winchester gallons, divide the displacement by 231; to reduce to English imperial gallons, divide the displacement by 277.27.*

Or,

$$D_p = L\,A\,N\,S,$$

$$D_c = \frac{L\,A\,N}{1{,}728},$$

$$D_{wg} = \frac{L\,A\,N}{231},$$

$$D_{eg} = \frac{L\,A\,N}{277.27},$$

where

L = length of stroke in inches;
A = area of piston or plunger in square inches;
N = number of delivery strokes per minute;
S = weight in pounds of a cubic inch of the liquid,
D_p = displacement in pounds per minute;
D_c = displacement in cubic feet per minute;
D_{wg} = displacement in Winchester gallons per minute;
D_{eg} = displacement in English imperial gallons per minute.

3. Attention is here called to the fact that there are three different gallons in use, of which the Winchester, or wine, gallon, measuring 231 cubic inches, is most commonly

used in America. In Great Britain and her colonies the imperial gallon, holding 277.27 cubic inches, is largely used as a measure. In most English-speaking countries the beer or ale gallon of 282 cubic inches capacity is also used, but almost exclusively for measuring the liquids mentioned. When the discharge of a pump is given in gallons in the United States of America, it is always understood, unless distinctly stated otherwise, to be in gallons measuring 231 cubic inches.

4. The **mean effective area** of the piston or plunger is equal to the area corresponding to the diameter only in case of outside-packed plunger pumps. In case of inside-packed and center-packed plunger pumps and double-acting piston pumps, the mean effective area is found by dividing the sum of the piston or plunger area and the same area diminished by the area of the piston rod by 2. Thus, in a double-acting inside-packed plunger pump having a plunger 10 inches in diameter and a 2-inch piston rod, the mean effective area is $\frac{10^2 \times .7854 + (10^2 \times .7854 - 2^2 \times .7854)}{2}$ $= 76.97$ square inches. In case of a single-acting piston pump, which generally is a lift pump, the effective area will be the area of the piston diminished by the area of the piston rod, since the piston rod is on the delivery side. In case of a differential pump having the plunger areas in the ratio of 1 to 2, the area of the smaller plunger is the effective area. In rough, approximate calculations of displacement, the correction for the area of the piston rod or plunger rod need not be made, and then the area of the piston or plunger is considered as the effective area. When the displacement requires to be accurately known, however, the mean effective area should be used.

EXAMPLE 1.—A single-acting plunger pump is driven by a crank whose radius is 8 inches and whose number of revolutions is 30 per minute. If the plunger is 6 inches in diameter, what is the displacement in cubic feet per minute?

SOLUTION.—The number of discharging strokes of the plunger is equal to the number of revolutions of the crank, or 30 per minute; the

length of the stroke is $8 \times 2 = 16$ inches. The area of the plunger is $6^2 \times .7854 = 28.27$ square inches. Applying rule **1**, we have

$$D_c = \frac{16 \times 28.27 \times 30}{1,728} = 7.85 \text{ cu. ft. per min. Ans.}$$

Example 2.—A center-packed double-acting duplex pump has plungers 24 inches diameter with 4-inch plunger rods. Each plunger makes 30 strokes per minute, the length of stroke being 32 inches. What is the displacement in American (Winchester) gallons per minute?

Solution —The mean effective area of the plungers is

$$\frac{24^2 \times .7854 + (24^2 \times .7854 - 4^2 \times .7854)}{2} = 446.1 \text{ square inches.}$$

Since the pump is duplex, there are $30 \times 2 = 60$ strokes per minute. Applying rule **1**, we get

$$D_{ag} = \frac{32 \times 446.1 \times 60}{231} = 3,707.8 \text{ gal. per min. Ans.}$$

DISCHARGE.

5. The **theoretical discharge** of a pump is equal to the *displacement.*

6. The **actual discharge** is generally less than the displacement, owing to leakage past the valves and piston and also to the return of water through the valves while they are in the act of closing.

SLIP.

7. The difference between the displacement and the actual discharge, expressed as a percentage of the displacement, is called the **slip** of a pump.

8. Negative Slip.—When the column of water in the suction and discharge pipes of a pump is long and the lift moderate, the energy imparted by the piston during the discharge stroke may be sufficient to keep the column in motion during all or a part of the return stroke. Under these conditions, the actual discharge may be greater than the displacement, and the slip is then said to be *negative.*

Rule 2.—*To calculate the slip of a pump, find the difference between the displacement and the actual discharge, multiply it by 100, and divide the product by the displacement. The quotient will be the slip expressed in per cent. of the displacement.*

Example.—A single-acting plunger pump with a plunger 8 inches in diameter and 36 inches stroke discharges 33.5 cubic feet of water per minute when making 35 discharging strokes. What is the slip?

Solution.—By rule **1**, the displacement is

$$\frac{36 \times 8^2 \times .7854 \times 35}{1,728} = 36.652 \text{ cubic feet per minute.}$$

By rule **2**, the slip is

$$\frac{(36.652 - 33.5) \times 100}{36.652} = 8.6 \text{ per cent., nearly. Ans.}$$

WORK DONE BY A PUMP.

9. The **useful work** in foot-pounds done by a pump is the product of the water raised in pounds multiplied by the vertical distance in feet from the surface of the water in the well or supply reservoir to the outflow end of the discharge pipe.

10. The **actual work** is always greater than the useful work. Force is required to overcome the friction of the piston or plunger in the cylinder or stuffingbox, and considerable force is also required to overcome the friction of the water in its passage through the pipes and the valves and passages of the pump. Some force must also be expended in giving the water the velocity it has when it leaves the discharge pipe.

The theoretical force required to drive the piston is equal to its area multiplied by the pressure due to a head equal to the vertical distance from the surface of the water in the well to the outlet of the discharge pipe. The actual force can be found by the aid of a pressure gauge or indicator attached to the pump cylinder, which will give the actual pressure on the piston in pounds per square inch.

According to the principles of hydraulics and the flow of water through pipes, it is evident that the power required to overcome the frictional resistance of the water will be reduced by making the pipes large and direct and the passages through the valves and pump of ample size and as direct as possible, so as to avoid loss from sudden change of direction of flow.

HORSEPOWER OF PUMPS.

11. The indicated horsepower developed in the cylinder or cylinders of a steam-driven or compressed-air-driven pump is found in exactly the same manner as with a steam engine and from the same data. The horsepower usefully expended is given by dividing the useful work done by the pump in 1 minute by 33,000. The ratio of the usefully expended horsepower to the indicated horsepower is an indication of the mechanical efficiency of the pumping apparatus considered as a whole.

12. It is often required to estimate what horsepower will be required to pump a given quantity of water per minute to a given elevation or against a given pressure. This problem can only be solved approximately by a general rule, there being a number of variable factors entering into the solution, such as the general run and length of the piping, the design of the water end, the degree of workmanship, etc. The influence of some of these factors cannot be determined beforehand with any great degree of accuracy, and for that reason any general rule for estimating the required horsepower must be based on a low mechanical efficiency of the pumping apparatus in order to leave an ample margin for safety.

13. In estimating upon the probable horsepower, it is occasionally necessary to convert a given pressure into a head of water in feet that will exert the same pressure. This can be readily done by multiplying the given pressure by 2.3.

14. If the volume of water to be discharged per minute is given in cubic feet and the vertical height from the suction level to the discharge level in feet is known, the foot-pounds of work to be done is $62.5 \times$ volume $\times$ vertical height, taking the weight of a cubic foot of water as 62.5 pounds. Consequently, the theoretical horsepower is

$$\frac{62.5 \times \text{volume} \times \text{vertical height}}{33,000},$$

or
$$\frac{\text{foot-pounds of work to be done}}{33,000}.$$

Assuming an efficiency of 70 per cent., the actual horsepower will be

$$\frac{100 \times \text{foot-pounds of work to be done}}{70 \times 33,000}$$

$$= \frac{\text{foot-pounds of work to be done}}{23,100}$$

Hence the following rule:

Rule 3.—*To estimate the probable horsepower required to drive a pump, multiply the weight to be discharged per minute by the vertical lift and divide by 23,100.*

Or,
$$H_e = \frac{WL}{23,100},$$

where H_e = estimated horsepower;
W = weight of water discharged per minute in pounds;
L = vertical lift in feet.

EXAMPLE.—About what horsepower will be required to discharge 350 gallons of water per minute, the total lift being 320 feet?

SOLUTION.—The weight of the Winchester, or ordinary American, gallon is 8.34 pounds, nearly. Hence, the weight of water to be pumped per minute is $350 \times 8.34 = 2,919$ pounds.

Applying rule **3**, we get

$$H_e = \frac{2,919 \times 320}{23,100} = 40 \text{ H. P., about.} \quad \text{Ans.}$$

15. When the weight of water to be discharged per minute and the pressure against which it is to be pumped are

known, the foot-pounds of work to be done is weight × pressure × 2.3. Assuming an efficiency of 70 per cent., the actual horsepower required is

$$\frac{100 \times \text{weight} \times \text{pressure} \times 2.3}{70 \times 33{,}000} = \frac{\text{weight} \times \text{pressure}}{10{,}043}.$$

Rule 4.—*To estimate the probable horsepower, multiply the weight of water to be pumped per minute by the pressure pumped against and divide by 10,043.*

Or, $$H_e = \frac{WP}{10{,}043},$$

where P = pressure per square inch;
W = weight of water per minute.

In rules **5**, **6**, **7**, and **8** the letters have the same meaning as in rules **3** and **4**.

Example.—A pump is to pump 400 cubic feet of water per hour against a pressure of 90 pounds per square inch. Estimate the probable horsepower required.

Solution.—Reducing the volume per hour to pounds per minute, we have

$$\frac{400 \times 62.5}{60} = 416.7, \text{ say } 417 \text{ pounds.}$$

Applying rule **4**, we get

$$H_e = \frac{417 \times 90}{10{,}043} = 3.1 \text{ H. P., about.} \quad \text{Ans.}$$

16. **Rule 5.**—*To estimate the vertical lift with a given horsepower, multiply the horsepower by 23,100 and divide by the weight of water to be delivered per minute.*

Or, $$L = \frac{23{,}100\, H_e}{W}.$$

Example.—A pump driven by a 10-horsepower engine is to discharge 2,000 pounds of water per minute. How high may this water be lifted, approximately?

Solution.—Applying rule **5**, we get

$$L = \frac{23{,}100 \times 10}{2{,}000} = 115.5 \text{ ft.} \quad \text{Ans.}$$

17. Rule 6.—*To estimate the probable discharge in pounds per minute, divide 23,100 times the horsepower by the vertical lift in feet.*

Or,
$$W = \frac{23{,}100\, H_e}{L}.$$

EXAMPLE.—How many pounds of water per minute, approximately, can a pump driven by a 25-horsepower engine discharge at a height of 42 feet?

SOLUTION.—Applying rule **6**, we get

$$W = \frac{23{,}100 \times 25}{42} = 13{,}750 \text{ lb., about.} \quad \text{Ans.}$$

18. Rule 7.—*To estimate the pressure that can be pumped against, multiply the horsepower by 10,043 and divide by the weight to be pumped per minute.*

Or,
$$P = \frac{10{,}043\, H_e}{W}.$$

EXAMPLE.—A 9-horsepower pump is to discharge 6,000 pounds of water per minute. Estimate against what pressure this can be discharged.

SOLUTION.—Applying rule **7**, we get

$$P = \frac{10\ 043 \times 9}{6{,}000} = 15 \text{ lb. per sq. in.} \quad \text{Ans.}$$

19. Rule 8.—*To estimate the probable discharge in pounds per minute, multiply the horsepower by 10,043 and divide by the pressure to be pumped against.*

Or,
$$W = \frac{10{,}043\, H_e}{P}.$$

EXAMPLE.—How much water may a pump be estimated to discharge in Winchester gallons per minute when the pump is 40-horsepower and pumps against a pressure of 100 pounds per square inch?

SOLUTION.—Applying rule **8**, we get

$$W = \frac{10{,}043 \times 40}{100} = 4{,}017.2 \text{ pounds per minute.}$$

Since a Winchester gallon weighs 8.34 pounds, we have

$$\frac{4{,}017.2}{8.34} = 481.7 \text{ gal. per min.} \quad \text{Ans.}$$

SIZE OF PISTONS AND PLUNGERS.

20. Before the size of a piston or plunger for the water end of a pump can be determined, the quantity of water to be pumped and the piston speed must be known. The piston speed is the number of feet traveled per minute by the plunger when *discharging* water; that is, it equals the length of the stroke in feet multiplied by the number of *working* strokes per minute. If the pump is double-acting, the number of working strokes is the same as the total number of plunger strokes, both forward and back; if single-acting, half that number. If the pump is duplex, it is advisable to consider only one side in determining the size of plunger or piston, designing it to suit one-half the total quantity of water to be delivered. In direct-acting steam pumps the piston speed is generally about 100 feet; at least it is customary to design them on this assumption, and then to run the pump faster or slower to suit the required delivery, opening or closing the throttle valve to vary the speed of the pump.

21. Knowing the actual volume of water to be discharged in 1 minute in cubic feet, the plunger or piston area in square feet will be $\frac{\text{discharge}}{\text{piston speed}}$, theoretically. But in practice the diameter of the plunger or piston is given in inches, hence the area should be expressed in square inches.

Then, $$\text{area} = \frac{\text{discharge in cubic feet} \times 144}{\text{piston speed in feet}},$$

and the corresponding diameter in inches will be

$$\sqrt{\frac{\text{discharge} \times 144}{.7854 \times \text{piston speed}}}.$$

22. Since there is always more or less slip of the water, it is usual to design the pump on the assumption that it must pump 1.25 times the actual amount of water. On this assumption the plunger or piston diameter in inches will be

$$\sqrt{\frac{\text{discharge} \times 1.25 \times 144}{.7854 \times \text{piston speed}}},$$

or $$\sqrt{\frac{229 \times \text{discharge}}{\text{piston speed}}}.$$

Rule 9.—*To find the diameter of a plunger or piston in inches, multiply the discharge in cubic feet per minute by 229 and divide the product by the piston speed in feet per minute. Extract the square root of the quotient.*

Or, $$d = \sqrt{\frac{229\,D}{S}},$$

where d = diameter of piston or plunger in inches;
D = actual discharge in cubic feet per minute;
S = piston speed.

When the discharge is given in pounds, gallons, or any other unit of volume, it should be reduced to cubic feet before applying rule **9**.

Example.—What should be the diameter of a pump plunger required to discharge 130 Winchester gallons per minute, the speed of the plunger being 90 feet per minute?

Solution.—Reducing the gallons to cubic feet, we have

$$\frac{130 \times 231}{1{,}728} = 17.378 \text{ cubic feet per minute.}$$

Applying rule **9**, we get

$$d = \sqrt{\frac{229 \times 17.378}{90}} = 6.65 \text{ in., nearly. Ans.}$$

23. Rule 10.—*To estimate the probable discharge in cubic feet, square the diameter of the plunger or piston in inches and multiply by the piston speed. Divide the product by 229.*

Or, $$D = \frac{d^2\,S}{229}.$$

Example.—How many pounds of water per hour may a duplex double-acting pump be expected to discharge when the diameter of the plungers is 6 inches, the length of stroke 24 inches, and each plunger makes 40 strokes per minute?

Solution.—The piston speed is $\frac{24}{12} \times 40 = 80$ feet per minute. The probable discharge per minute in cubic feet, by rule **10**, is

$$D = \frac{6^2 \times 80}{229};$$

$$\text{per hour, } D = \frac{6^2 \times 80 \times 60}{229}.$$

The discharge in pounds per hour, taking 62.5 pounds as the weight of a cubic foot of water, is

$$D = \frac{6^2 \times 80 \times 60 \times 62.5}{229}$$

for one side of the pump. For both sides,

$$D = \frac{6^2 \times 80 \times 60 \times 62.5 \times 2}{229} = 94,323 \text{ lb.} \quad \text{Ans.}$$

In applying rule **10** it is to be observed that the result will be less than given by multiplying the displacement per stroke by the number of strokes per minute, as called for by rule **1**. The reason for this discrepancy is obvious; rule **1** gives the theoretical discharge, while rule **10** gives about what the pump may actually be expected to discharge.

24. In direct-acting steam pumps the normal piston speed is generally 100 feet per minute. On this basis the probable discharge in cubic feet, by rule **10**, is $D = \frac{d^2 \times 100}{229}$, and in Winchester gallons the discharge is $\frac{d^2 \times 100 \times 1,728}{231 \times 229} = 3.26\, d^2$.

Rule 11.—*To roughly approximate the probable normal discharge of a direct-acting steam pump in gallons, multiply the square of the diameter of the plunger or piston by 3.26.*

Or, $$D_g = 3.26\, d^2,$$

where D_g = discharge in gallons per minute;
d = diameter of piston or plunger in inches.

25. The theoretical normal discharge in gallons per minute at a piston speed of 100 feet is given almost exactly by multiplying the square of the diameter of the plunger or piston by 4. For a duplex pump the discharge is double that given by rule **11**.

EXAMPLE.—What may the discharge in gallons of a duplex pump with 6-inch plungers be roughly estimated at?

SOLUTION.—Applying rule **11**, we get $D_g = 3.26 \times 6^2$ for each side, or

$$D_g = 3.26 \times 6^2 \times 2 = 235 \text{ gal. per min.} \quad \text{Ans.}$$

26. Having determined the proper plunger or piston diameter for the chosen piston speed, it remains to choose either a length of stroke or a number of strokes in order to determine either the number of strokes or the length of stroke. The ratio of the diameter to the length of stroke varies between very wide limits in practice, being as low as 1 : 1 and as high as 1 : 5. Obviously, the greater the ratio, the fewer times will the valves have to be moved, hence a great ratio is generally chosen for pumps that have to run continuously in a hard, rough service. Having chosen a length of stroke, use the following rule:

Rule 12.—*To find the number of strokes, divide the piston speed in feet by the chosen length of stroke in feet. To find the length of stroke in feet, divide the piston speed in feet by the number of delivery strokes per minute.*

Or,

$$N = \frac{P}{L},$$

and

$$L = \frac{P}{N},$$

where P = piston speed;
N = number of delivery strokes per minute;
L = length of stroke in feet.

EXAMPLE.—What should be the length of stroke for a piston speed of 100 feet if the number of strokes per minute is 40?

SOLUTION.—Applying rule **12**, we get

$$L = \tfrac{100}{40} = 2.5 \text{ ft.},$$

or

$$2.5 \times 12 = 30 \text{ in. Ans.}$$

SIZE OF STEAM END.

27. In a direct-acting steam pump the size of the steam-end cylinder depends on two factors, which are the steam pressure available and the resistance against which the pump is to force the water. The stroke of the steam piston and water piston obviously are the same, both being rigidly connected to the same rod.

28. The forces acting on the steam piston and water piston are equal when the area of the steam piston × the steam pressure = area of water piston × pressure pumped against. But in order that there may be an ample margin to overcome the frictional resistances, which make the actual resistance to the motion of the water piston greater and lessen the force that impels the steam piston forwards, the area of the steam piston should be, at least, 40 per cent. in excess of its theoretical area. On this basis, we have area of steam piston

$$= \frac{1.4 \times \text{area of water piston} \times \text{pressure}}{\text{steam pressure}},$$

and diameter of steam piston

$$= \sqrt{\frac{1.4 \times \text{area of water piston} \times \text{pressure}}{.7854 \times \text{steam pressure}}},$$

or diameter of steam piston

$$= \sqrt{\frac{1.8 \times \text{area of water piston} \times \text{pressure}}{\text{steam pressure}}}.$$

Rule 13.—*To find the minimum diameter of the steam piston of a direct-acting steam pump, multiply 1.8 times the area of the water piston in square inches by the pressure in pounds per square inch to be pumped against; divide by the available steam pressure and extract the square root of the quotient.*

Or, $$d_m = \sqrt{\frac{1.8\, a\, p}{P}},$$

where d_m = minimum diameter of steam piston in inches;
a = area of water piston;
p = pressure to be pumped against;
P = steam pressure available.

Example.—What should be the minimum diameter of the steam piston for a pump having a plunger 8 inches in diameter, the available steam pressure being 75 pounds per square inch and the water to be pumped against a pressure of 200 pounds per square inch?

SOLUTION.—The area of the plunger is $8^2 \times .7854 = 50.27$ square inches. Applying now rule **13**, we get

$$d_m = \sqrt{\frac{1.8 \times 50.27 \times 200}{75}} = 15.5 \text{ in. Ans.}$$

29. It is to be observed that rule **13** applies equally well to steam- and air-driven pumps. It can also be applied to simple pumps of the crank-and-flywheel type using steam expansively. In the latter case, the mean effective pressure throughout the stroke must be taken as the available steam pressure. Rule **13** is especially useful in deciding whèther a given pump will pump against a known pressure with the existing sizes of steam and water pistons. It will also be found very useful in selecting a pump for a given service from the catalogues of manufacturers.

30. In boiler-feed pumps the steam pressure available and the pressure pumped against are practically equal, so that it might be expected that the area of the steam piston would be made about 40 per cent. larger than the area of the water piston. In actual practice it is found, however, that pump manufacturers prefer to make the steam piston about 3 times the area of the water piston in very small pumps and about twice the area of the water piston in large pumps. The steam piston of boiler-feed pumps is made so largely in excess of what it really needs to be merely as a matter of safety; its large size simply tends to insure a prompt starting of the pump under almost all conditions likely to arise in practice.

31. The steam end of direct-acting pumps and of direct-connected crank-and-flywheel pumps, where the steam and water pistons move together, is rarely proportioned on the basis of horsepower required to do the work, it being much easier to calculate the size of the steam end by rule **13**.

32. When a power pump is driven by a separate steam engine, through the intervention of belting or gearing, the engine itself is generally selected on the basis of horsepower

required to do the work, and then the question as to what size of engine to use presents itself. This problem is capable of an infinite number of solutions, since a variation of either of the two factors—piston speed and mean effective pressure—will cause a difference in size. In general, the steam engine for driving a pump is selected in exactly the same manner, as far as its size is concerned, as a steam engine for any other service.

THE DUTY OF STEAM PUMPS.

DEFINITION.

33. The ratio between the work done by a pump and a certain amount of coal, steam, or heat units used to do the work is called the **duty** of the pump.

During a certain time, say an hour or a day, the pump will raise a quantity of water through a certain height and thus perform a definite amount of work. To do this work, the pump has received from the boilers a certain number of heat units or a number of pounds of steam; or, if the boilers are included as a part of the system, the work has been accomplished by consuming a certain amount of coal. The pump is credited with the work it has performed in the stated time and is charged with the number of heat units, pounds of steam, or pounds of coal it has used in doing the work. It is plain that the economy of the pump or pumping engine is measured by the ratio of the work performed to the steam consumed or the coal burned. Thus, if one pump does 50,000,000 foot-pounds of work with a coal consumption of 100 pounds and another under the same conditions does 36,000,000 foot-pounds and consumes only 60 pounds of coal, the latter is evidently the more economical, since the ratio of work to coal consumption is larger, being $(36{,}000{,}000 \div 60) \times 100 = 60{,}000{,}000$ foot-pounds of work with a coal consumption of 100 pounds.

DUTY BASED ON COAL CONSUMPTION.

34. When the duty is based on the consumption of coal, it is customary to assume 100 pounds of coal as the fuel unit; that is, the duty is defined as the number of foot-pounds of work performed for each 100 pounds of coal burned. Then,

$$\text{Duty} = \text{foot-pounds of work} \div \frac{\text{pounds of coal}}{100},$$

or,

$$\text{Duty} = \frac{\text{foot-pounds of work} \times 100}{\text{pounds of coal}}.$$

Rule 14.—*To find the duty of a pump per 100 pounds of coal, multiply together 100, the weight of water pumped in a given time in pounds, and the vertical distance in feet from the level of supply to the level of discharge. Divide the product by the coal consumption in the same time in pounds.*

Or,

$$D = \frac{100\, w\, h}{W},$$

where

D = duty;
w = weight of water in pounds;
W = weight of coal in pounds;
h = vertical lift in feet.

EXAMPLE.—A pump raises 130,000 pounds of water 60 feet and the operation requires the combustion of 25 pounds of coal. What is the duty?

SOLUTION.—Applying rule **14**, we have

$$D = \frac{100 \times 130{,}000 \times 60}{25} = 31{,}200{,}000 \text{ ft.-lb. per 100 lb. of coal. Ans.}$$

35. The duty based on the coal consumption is of practical value, as it gives an idea of the coal required by a pump of a given type for the performance of a stated quantity of work. It is clear, however, that if a comparison of the merits of two pumps is to be made, the coal must be of the same quality in each case. Further, the boilers supplying steam to the pumps should be of the same type or at least have the same evaporative capacity. This is a point of great importance. One hundred pounds of good bituminous or anthracite coal may, under favorable conditions, evaporate

1,000 to 1,100 pounds of water; that is, furnish that number of pounds of steam to the pump. In many cases, however, the 100 pounds of coal, if of inferior quality and burned under a poor boiler, will not furnish the pump more than 450 to 600 pounds of steam. Under such conditions the duty of the pump based on the coal consumption would not be a fair indication of its efficiency and would not serve as a satisfactory basis for comparing it with other pumps.

DUTY BASED ON STEAM CONSUMPTION.

36. In order to avoid the drawbacks incidental to basing the duty of pump on the coal consumption, it is the custom of some pump makers to specify that the coal consumption shall be estimated on the supposition that a pound of coal evaporates 10 pounds of water, or, in other words, furnishes 10 pounds of steam to the pump. To make this clear, suppose that in a duty trial 32,000 pounds of steam were used by the pump; the duty of the pump would be calculated on the assumption that the coal consumption was $32{,}000 \div 10 = 3{,}200$ pounds, though 5,000 pounds might actually have been used. If 1 pound of coal is assumed to furnish 10 pounds of steam, 100 pounds of coal will furnish 1,000 pounds of steam; hence, the duty based on steam consumption may be defined as the number of foot-pounds of work done by the pump per 1,000 pounds of dry steam supplied it. Then,

$$\text{Duty} = \frac{\text{foot-pounds of work} \times 1{,}000}{\text{pounds of steam}}.$$

Rule 15.—*To find the duty of a pump per 1,000 pounds of dry steam, multiply together the weight of water pumped in pounds, the vertical distance in feet from the level of supply to the level of discharge, and 1,000. Divide by the weight of steam supplied in pounds.*

Or,
$$D = \frac{1{,}000\, w\, h}{S}$$

where S = weight of dry steam supplied in pounds and the other letters have the same meaning as in rule **14**.

EXAMPLE.—A pump lifted 7,920,000 pounds of water 126 feet with 8,100 pounds of steam. What is its duty?

SOLUTION.—Applying rule **15**, we get

$$D = \frac{1,000 \times 7,920,000 \times 126}{8,100}$$

$= 123,200,000$ ft.-lb. of work per 1,000 lb. of dry steam. Ans.

37. The basis of 1,000 pounds of dry steam is more scientific and better adapted for duty trials than that of 100 pounds of coal, but it is open, nevertheless, to objections. Not only is it difficult to determine the exact weight of dry steam entering the pump, but also 1,000 pounds of steam at 160 pounds pressure will do more work in the cylinder than 1,000 pounds of steam at 60 pounds pressure. If scientific accuracy is sought, the pressure of the steam should be specified in addition to the weight.

DUTY BASED ON HEAT UNITS SUPPLIED.

38. On account of the objections to the basis of comparison then used, a committee of The American Society of Mechanical Engineers in 1891 recommended a new basis for the estimation of duty. Whether the furnace consumes 100 or 200 pounds of coal, whether the steam is at 60 or 160 pounds pressure, wet or dry, the steam cylinders of the pump or pumping engine receive in a given time a definite number of British thermal units. We have seen that if each of two pumps is allowed 100 pounds of coal to do a certain amount of work, one of the pumps may be at a disadvantage on account of the poor quality of the coal or the inefficiency of the boiler. If each is allowed 1,000 pounds of dry steam, there may be an inequality because of a difference in the steam pressure in the two cases. If, however, each pump is furnished with an equal number of heat units, each has exactly the same stock in trade, and the merit of each pump can be gauged by the use it makes of the heat units furnished it, that is, by the ratio of the work performed to number of heat units supplied.

39. If a pound of water has a temperature of 212°, it requires 966.1 B. T. U. to change it to steam at atmospheric pressure. If the water has originally a lower temperature or is converted into steam at higher pressure, more B. T. U. are required to accomplish the change. Roughly speaking, if the temperature of the feed and pressure of the steam are not given, about 1,000 to 1,100 B. T. U. are equivalent to a pound of steam. Therefore, 1,000 pounds of steam are equivalent to about $1,000 \times 1,000 = 1,000,000$ B. T. U.

40. Looking at the question in another light, a pound of good coal when burned produces about 13,500 to 14,000 B. T. U. by the combustion. A boiler of fairly good efficiency will utilize perhaps 10,000 of these 13,500 B. T. U., the rest being lost by radiation, in the production of chimney draft, and in other ways. From 100 pounds of coal the boiler is able to extract $100 \times 10,000 = 1,000,000$ B.T.U., which are eventually given up to the pump. It thus appears that 100 pounds of coal and 1,000 pounds of steam are each approximately equivalent to 1,000,000 B. T. U.; for this reason, the committee of The American Society of Mechanical Engineers recommended that the new basis for estimating duty should be 1,000,000 B. T. U.

41. The heat-unit basis is now very extensively used and is recommended in preference to the others. It may be expressed as follows:

The duty of a pumping engine is equal to the total number of foot-pounds of work actually done by the pump divided by the total number of heat units in the steam used by the pump, and this quotient multiplied by 1,000,000. The heat units in the steam used for driving the auxiliary machinery, such as the air pump and circulating pump of the condenser, if one is used, and the boiler-feed pumps are charged as heat units supplied to the pump.

42. The number of foot-pounds of work done by the pump is to be found as follows: A pressure gauge is attached to the discharge pipe and a vacuum gauge to the

suction pipe, both as near the pump as convenient; then the net pressure against which the pump plunger works is equal to the sum or difference in the pressures shown by these two gauges increased by the hydrostatic pressure due to the difference in level of the points in the pipes to which they are attached. In case the gauge in the suction pipe indicates a vacuum, the sum of the pressures indicated by the gauges is taken, but when the water flows into the suction pipe under a head, so that the suction gauge indicates a pressure above the atmospheric pressure, the difference in the two pressures indicated by the gauges is taken.

43. The number of foot-pounds of work done during the trial is equal to the continued product of the net area of the plunger in square inches (making allowance for piston rods), the length of the plunger stroke in feet, the number of plunger strokes made during the trial, and the net pressure in pounds per square inch against which the plungers work.

44. The pressure corresponding to the vacuum in inches indicated by the gauge on the suction pipe is found by multiplying the gauge reading in inches by .4914, and the pressure corresponding to the difference in the level of the two gauges by multiplying this difference in feet by .434. The number of heat units furnished to the pump is the number of British thermal units in the steam from the boilers and is to be determined by an evaporation test of the boilers.

Rule 16.—*To determine the duty of a pump per 1,000,000 B. T. U., multiply the net pressure against which the plunger works, in pounds per square inch, by the net area of the plunger in square inches, by the average length of stroke in feet, the total number of delivery strokes made during the trial, and by 1,000,000. Divide the product by the total number of B. T. U. supplied during the trial.*

Or, $$D = \frac{1{,}000{,}000\,(P \pm p + S)\,A\,L\,N}{H},$$

where D = duty;

P = pressure in pounds per square inch in the discharge pipe;

p = pressure in pounds per square inch in the suction pipe, to be added in case of a vacuum and to be subtracted in case of pressure above atmospheric pressure in the suction pipe;

S = pressure in pounds per square inch corresponding to difference in level between the gauges;

A = average effective area of plunger in square inches;

L = length of stroke of pump plunger in feet;

N = total number of delivery strokes;

H = total number of B. T. U. supplied.

EXAMPLE.—A crank-and-flywheel pump has two double-acting water plungers, each 20 inches in diameter and 36 inches stroke. Each plunger has a piston rod 3 inches in diameter extending through one pump-cylinder head.

During a 10-hour duty trial the total heat in the steam supplied to the engine was 35,752,340 B. T. U. and the engine made 9,527 revolutions. If the average pressure indicated by a gauge on the discharge pipe was $95\frac{1}{2}$ pounds, the average vacuum indicated by a gauge on the suction pipe $8\frac{1}{4}$ inches, and the difference in level between the centers of the vacuum and the pressure gauge 8 feet, what was the duty?

SOLUTION.—The area of a plunger 20 inches in diameter is 314.16 square inches and the cross-sectional area of a rod 3 inches in diameter is 7.07 square inches. Since the rod extends through only one end of the pump cylinder, the average effective area of the two ends of each plunger is $314.16 - \frac{7.07}{2} = 310.63$ square inches.

The pressure corresponding to a vacuum of $8\frac{1}{4}$ inches is $p = 8.25 \times .4914 = 4.05$ pounds per square inch, and the pressure corresponding to a difference in level of 8 feet is $S = 8 \times .434 = 3.47$ pounds per square inch.

Since there are two double-acting plungers, the total number of plunger strokes corresponding to 9,527 revolutions is $N = 9{,}527 \times 4$ 38,108

Applying rule **16**, we get

$$D = \frac{1{,}000{,}000 \times (95.5 + 4.05 + 3.47) \times 310.63 \times 3 \times 38{,}108}{35{,}752{,}340}$$

$$= 102{,}328{,}800 \text{ ft.-lb. per } 1{,}000{,}000 \text{ B. T. U.} \quad \text{Ans.}$$

DUTY BASED ON VOLUME OR WEIGHT PUMPED.

45. In large pumping plants it often happens that the pressure pumped against is either constant or practically so. In such a plant a record is often kept for the purpose of comparing the performance of the plant from week to week or month to month with its former performances. The records may be kept in number of gallons pumped per pound of coal; in cubic feet pumped per pound of coal; in weight of water in pounds or tons pumped per pound of coal; or the record may be kept per ton or bushel of coal, etc. Duty computed on such a basis is spoken of as **gallon duty, cubic-foot duty, pound duty, ton duty,** etc.; while such a duty is very valuable in showing variations in efficiency of a given plant at different times, it cannot be used as a basis of comparison between the performances of different pumping plants, and when so used will be utterly misleading.

Instead of keeping the records in terms of quantity of water pumped per pound of coal, they may advantageously be kept in terms of water pumped per dollar; the records then show variations in efficiency in their true light.

EXPRESSING THE DUTY OF A PUMP.

46. The question "in what terms shall the duty of a pump be expressed" depends for its answer on the purpose for which it is required that the duty be known. If the duty is merely required to be known in order that the performances of a given pump at different times may be compared with one another, the duty may be based on coal consumption, steam consumption, or volume pumped per some unit of fuel or money. If, however, the performance of a pump is to be compared with that of others working probably under entirely different conditions, the foot-pounds of work done per 1,000,000 B. T. U. is the only true basis of comparison.

AVERAGE DUTIES.

47. Small direct-acting pumps for general service have a duty of 15,000,000 foot-pounds per 1,000 pounds of steam used. Compound direct-acting pumps of 5,000,000 gallons capacity in 24 hours should give a duty of 50,000,000 foot-pounds per 1,000 pounds of steam used. Large municipal pumping engines of 20,000,000 gallons capacity in 24 hours have given a duty of 160,000,000 foot-pounds per 1,000 pounds of dry steam used by the engine.

Centrifugal and rotary pumps have a duty depending on the type of engine used to drive them, and since they usually run at high speed and the conditions for economical performance are good, an economical type of engine can be used and the duty of the combined unit thus made to compare very favorably with that of the reciprocating pump.

48. Tests of the duty of pumps and pumping engines have generally been made when the machinery was in first-class condition. It is customary to run these machines from 6 months to 1 year after they are installed before making the test, the object being to bring all the journals into a good bearing condition; also, the piston and all the other rubbing surfaces will be much improved by the polishing and the working of oil into the pores of the iron during running. These high duties can only be maintained by the closest attention to every detail by the operating engineer. Indicator cards should be taken from both the steam and the water ends of the pumps every week and closely compared with previous indications to see that the highest state of efficiency is being maintained within the working parts of the pump.

EFFICIENCY OF VARIOUS TYPES OF PUMPS.

49. When the **efficiency of a pump** is spoken of, its *mechanical* efficiency is generally meant, unless stated otherwise. This is measured by dividing the actual or net horsepower of the machine by its indicated horsepower, and the quotient, when multiplied by 100, will be the efficiency

expressed in per cent. Very small direct-acting steam pumps have an efficiency of about 50 per cent., the efficiency increasing with the size of the pump up to about 80 per cent. The efficiency of direct-acting steam pumps and also of pumps in general increases with the size by reason of the decrease in the ratio that the frictional resistances bear to the indicated horsepower as the size of the pipes and passages is increased. The reason that the frictional resistances decrease can readily be seen when it is considered that by doubling the diameter of a pipe and keeping the velocity of flow the same, the discharge will be increased four times, while the surface that the water is rubbing against is only doubled.

50. Large vertical municipal pumping engines have shown an efficiency as high as 96 per cent.; horizontal medium-size crank-and-flywheel pumps show efficiencies as high as 90 per cent. The efficiency of centrifugal, rotary, and screw pumps varies between 40 and 66 per cent., about, depending on the size; small pumps are less efficient than larger ones. This efficiency of centrifugal, rotary, and screw pumps is the *efficiency of the pump itself*, and not the combined efficiency of the pump and engine, or motor, driving it.

SIZE OF SUCTION AND DELIVERY PIPES.

51. Experience has demonstrated that for satisfactory work the flow of water in the suction pipes of pumps should not exceed 200 feet per minute, and it should not be more than 500 feet in the delivery pipe for a duplex double-acting pump, or 400 feet for a single-cylinder double-acting pump.

Knowing the volume of water that is to flow through or to be discharged from a pipe in 1 minute, the area of the suction and delivery pipes can readily be determined.

The volume of water in cubic feet discharged from a pipe in 1 minute is equal to the velocity in feet per minute times the area of the pipe in square feet. Then,

$$\text{the area of the pipe} = \frac{\text{volume in cubic feet per minute}}{\text{velocity in feet per minute}}.$$

As there are 144 square inches in a square foot,

$$\text{the area of the pipe in square inches} = \frac{144 \times \text{volume in cubic feet per minute}}{\text{velocity in feet per minute}}.$$

Rule 17.—*To find the area of a pipe in square inches to discharge a given volume of water per minute, divide the product of the volume in cubic feet and 144 by the allowable velocity in feet per minute.*

Or, $$A = \frac{144\,V}{v},$$

where A = area of pipe in square inches;
V = volume to be discharged per minute;
v = allowable velocity.

When the weight of water is given in pounds, divide it by 62.5 to reduce it to cubic feet; when the volume is given in Winchester gallons, divide it by 7.48 to reduce it to cubic feet.

EXAMPLE.—What should be the areas of the suction and delivery pipes for a single double-acting pump that is to discharge 6,250 pounds of water per minute?

SOLUTION.—Reducing the weight to cubic feet, we have $\frac{6,250}{62.5} = 100$ cubic feet. Then, applying rule **17**, we have

$$A = \frac{144 \times 100}{200} = 72 \text{ square inches}$$

as the area of the suction pipe, and

$$A = \frac{144 \times 100}{400} = 36 \text{ square inches}$$

as the area of the delivery pipe. The nearest standard nominal sizes of pipe to be used would be 10-inch and 7-inch. Ans.

52. The velocity with which water will flow through the delivery pipe of a pump when the area of the water cylinder, the area of the delivery pipe, and the piston speed of the pump are known, is given by the following rule:

Rule 18.—*Multiply the area of the water piston by the piston speed and divide this product by the area of the delivery pipe.*

Or, $$v = \frac{a\,S}{A},$$

where v = velocity in feet per minute;
A = area of delivery pipe in square inches;
a = area of water piston in square inches;
S = piston speed in feet per minute.

EXAMPLE.—If the water piston of a pump has an area of 12 square inches and moves at a speed of 100 feet per minute, what will be the velocity of the water in the delivery pipe if the latter has an area of 2 square inches?

SOLUTION.—Applying rule **18**, we get

$$v = \frac{12 \times 100}{2} = 600 \text{ ft. per min.} \quad \text{Ans.}$$

EXAMPLES FOR PRACTICE.

1. The plungers of a center-packed double-acting duplex pump are 20 inches in diameter and the plunger rods are 3½ inches in diameter. Each plunger makes 45 strokes per minute, the length of stroke being 24 inches. What is the displacement in cubic feet per minute?
Ans. 386.69 cu. ft.

2. In the above example, if the pump delivers but 360 cubic feet per minute, what is the slip? Ans. 6.9 per cent.

3. Approximately, what horsepower will be required to deliver 60 cubic feet of water per minute, the total lift being 470 feet?
Ans. 76.3 H. P.

4. What is the probable horsepower required to deliver 3,500 gallons of water per hour against a pressure of 115 pounds per square inch?
Ans. 5.57 H. P.

5. A pump driven by a 25-horsepower engine is to discharge 60 cubic feet of water per minute. How high may this water be lifted, approximately? Ans. 154 ft.

6. Approximately, how many gallons of water per hour can a pump driven by a 30-horsepower engine deliver at a height of 65 feet?
Ans. 1,278.3 gallons.

7. Approximately, against what pressure can a 20-horsepower pump discharge 2,500 cubic feet of water per hour?
Ans. 77 lb. per sq. in.

8. About how many cubic feet of water per minute may a 75-horsepower pump be expected to discharge against a pressure of 150 pounds per square inch? Ans. 80.3 cu. ft. per min.

9. A pump is required to discharge 1,800 cubic feet of water per hour. If the speed of the plunger is 100 feet per minute, what should be the diameter of the plunger? Ans. 8.29 in., nearly.

10. If the plunger of a double-acting pump is 10 inches in diameter and the length of stroke is 24 inches, how many gallons of water per hour may the pump be expected to deliver if it makes 45 strokes per minute? Ans. 17,639 gal. per hr.

11. Roughly estimate the discharge in gallons of a direct-acting steam pump having a plunger 7 inches in diameter.

Ans. 159.7 gal. per min.

12. If the piston speed is 90 feet per minute and the length of stroke 2 feet, how many strokes per minute will the pump make? Ans. 45.

13. Calculate the minimum diameter of the steam piston for a pump having a plunger 12 inches in diameter, the pressure to be pumped against being 175 pounds per square inch and the available steam pressure 100 pounds per square inch. Ans. 18.87 in.

14. What is the duty per 100 pounds of coal of a pump that raises 330,000 pounds of water 125 feet and requires 110 pounds of coal to perform the operation? Ans. 37,500,000 ft.-lb.

15. If 20,016 pounds of steam are consumed by a pump in lifting 1,200,000 gallons of water 150 feet, what is the duty per 1,000 pounds of dry steam? Ans. 75,000,000 ft.-lb.

16. A double-acting pump has a stroke of 40 inches; the diameter of the plunger is 24 inches and the diameter of the piston rod, which extends through one pump-cylinder head, is 3½ inches. During a 12-hour duty trial the total heat supplied to the engine was 47,652,500 B. T. U. and the engine made 23,200 strokes. What was the duty of the pump per 1,000,000 B. T. U. if the average pressure indicated by the gauge on the discharge pipe was 122 pounds, the average vacuum indicated by a gauge on the suction pipe 5 inches, and the difference in level between the centers of the vacuum gauge and pressure gauge was 10 feet?

Ans. 93,555,123 ft.-lb.

17. Calculate the area of the suction and delivery pipes for a single-acting pump that is to discharge 1,250 gallons of water per minute.

Ans. $\begin{cases} \text{Suction pipe, 120.3 sq. in.} \\ \text{Delivery pipe, 60.15 sq. in.} \end{cases}$

18. If the plunger of a single-acting pump has a speed of 85 feet per minute and a diameter of 6 inches, what will be the velocity of the water in the delivery pipe if the latter has an area of 6 square inches?

Ans. 400.5 ft. per min.

SELECTION OF PUMPS.

SERVICE OF DIFFERENT TYPES OF PUMPS.

53. Introduction.—The service for which a pump is required determines its general type, that is, whether it is to be a plunger pump, a rotary pump, a centrifugal pump, or a screw pump.

54. Reciprocating Pumps.—The various types of reciprocating pumps are selected when high efficiency is required and a fluid for which they are suited is to be pumped.

55. Rotary Pumps.—The rotary pump is chosen when the fluid to be pumped is water holding in suspension large masses of soft material. It is much used in paper mills for pumping the pulp from one stage of its manufacture to another. Rotary pumps are small and occupy, relatively, but little space for their capacity; they are also light, simple, and inexpensive, but are low in efficiency and are short lived, particularly if the material pumped contains much sand or other grit. The rotary pump is used with good success on some steam fire-engines, where light weight and simplicity are more important than high efficiency.

56. Centrifugal Pumps.—Centrifugal pumps are used where large volumes of water are to be lifted to moderate heights. They are also well adapted for pumping large quantities of dirty water, and, hence, are also much used for dredging and for sewage pumping. The efficiency of the centrifugal pump is low, but it is extremely simple and occupies comparatively little space for its capacity. Like the rotary pump, it has no valves and the flow is continuous. It is less affected by sand and grit than is the rotary pump. Neither the rotary pump nor the centrifugal pump requires much, if any, foundation.

57. Displacement Pumps.—Under the head of displacement pumps may be classed the pulsometer, which has

no running parts. This type of pump is well adapted for pumping all kinds of gritty water and is used for sinking and contractor's purposes. It is very simple in construction, low in first cost, and is not liable to get out of order. The class of pumps known as air lifts are principally used for artesian-well service; they require an air compressor for operation, but the apparatus itself is simple and low in first cost.

58. Screw Pumps.—Screw pumps are adapted for the handling of thick liquids, such as hot tar, pitch, paraffin, soap, etc. They have a uniform discharge and occupy small space; a much higher efficiency is claimed for them than for rotary or centrifugal pumps.

RECIPROCATING PUMPS.

CLASSIFICATION.

59. The reciprocating pump is, in general, the most efficient and hence the most common pump. It is built in a large variety of designs to suit different conditions and varies in size between very wide limits. Reciprocating pumps may be classified in accordance with the service for which they are intended as boiler-feed pumps, general-service pumps, tank or light-service pumps, fire pumps, low steam-pressure pumps, pressure pumps, mine pumps, sinking pumps, ballast pumps, wrecking pumps, deep-well pumps, sewage pumps, vacuum pumps, power pumps, municipal pumping engines, etc.

BOILER-FEED PUMPS.

60. Boiler-feed pumps are used for supplying steam boilers with their necessary water supply. For low pressures they are usually made of the piston pattern or the inside-packed plunger patterns. The cylinders are generally brass lined; the valves are brass or hard composition, with

composition springs and guards, and the pumps, hence, are suitable for handling hot water. For pressures above 135 pounds the outside-packed plunger type is preferred. Boiler-feed pumps are made both vertical and horizontal and for pressures from 50 pounds to 300 pounds per square inch. They vary in size from those having water plungers 1 inch in diameter to those having plungers 10 inches in diameter. The single-cylinder type is much used for boiler feeding, but, perhaps undeservedly, they have not the reputation for continuous action under all circumstances that is given to the duplex pump. Power pumps are often used for boiler feeding.

61. Whenever possible the boiler feeding apparatus should be in duplicate, so that the stoppage of one set will not affect the running of the plant. This end is generally secured by installing both a pump and an injector, each having a capacity sufficient for the needs of the plant.

62. Steam-driven crank-and-flywheel pumps are occasionally used, but they are open to the serious objection that they cannot always be run slow enough to suit the demand without stopping on the centers. In very large electrical installations, the electrically driven power pump is the most economical and satisfactory arrangement. Mills and factories often use the two-throw power pump having a movable crankpin, by means of which the stroke and hence the quantity of water pumped can be adjusted to suit the requirements. By this means a constant supply of feedwater equal to the demands for steam can be obtained, which is superior to the practice of pumping large quantities of water into the boilers at intervals. Boiler-feed pumps should not be required to run faster than 100 feet per minute piston speed. The velocity of water through the suction pipe should not exceed 200 feet and through the delivery should not be more than 400 feet. If the pipes are long or fitted with elbows, the velocity should be correspondingly decreased.

63. In determining the proper capacity of a pump for boiler feeding, the pump should be selected in reference to the amount of steam the boilers must supply. This is rarely only the amount used by the engine; in fact, in many industrial establishments much more steam is needed for other machinery than for the engine. Hence, it is best to always base the estimate as to the amount of water required on the maximum capacity of the boilers.

64. The maximum water consumption may be estimated in pounds per minute by one of the following rules, which hold good for average practice under natural draft. It will be observed that no rule based on the so-called "boiler horsepower" is given, for the reason that this is too variable a quantity to place any reliance on.

Rule 19.—*For plain cylindrical boilers multiply the product of the length and diameter in feet by .18.*

Rule 20.—*For tubular boilers multiply the heating surface in square feet by .06.*

Rule 21.—*Multiply the grate surface in square feet by 1.7.*

Rule 22.—*Multiply the estimated coal consumption in pounds per hour by .17.*

65. Whenever possible the feed-pumps should be located in the boiler room, so as to be directly in sight and in charge of the boiler attendant. In very large installations it is common to arrange the pumps in a separate pump house, they being then in charge of one of the assistant engineers, the boiler attendants regulating the supply to each battery by valves in the feedpipes.

GENERAL-SERVICE PUMPS.

66. General-service pumps are a line of pumps placed on the market by many of the pump builders to be used for any service where the water pressure does not exceed 150 pounds. They are generally of the plunger type and are built in sizes varying from those having a 4-inch to those

having a 16-inch plunger, and of a capacity varying from 100 gallons to 2,500 gallons per minute. They may be used for any service such as boiler feeding, fire, hydraulic elevator, or anywhere where the pressure to be pumped against is not greater than the limit stated.

TANK OR LIGHT-SERVICE PUMPS.

67. Tank or light-service pumps are of the same general form and interior construction as general-service pumps, except that the plungers are much larger in proportion to the steam cylinders, equalling or exceeding them in diameter. Such pumps cannot be used to feed their own boilers, but they are sometimes fitted with an attached pump for this purpose. Light-service pumps are commonly built of the same capacity as general-service pumps, but can only pump against low pressures.

FIRE PUMPS.

68. Fire pumps are most frequently of the duplex double-acting type with a ratio of area of steam cylinder to water piston of about 4 to 1. The duplex engine is chosen for this service on account of its simplicity and the peculiar adaptability of its motion to the high speed that is sometimes required in this service. A fire pump is frequently fitted up with a number of nozzles for hose connection. It should have relief valves, air and vacuum chambers of large capacity, steam and water gauges, priming pipes, and all the necessary valves.

Fire pumps, as implied by the name, are intended for use in case of fire, and are required to throw a large volume of water at high pressure.

LOW-PRESSURE STEAM PUMPS.

69. Low-pressure steam pumps are pumps intended for localities where only a low steam pressure is available, as in apartment houses, public and private buildings, etc.,

in which the pressure at which the steam heating system is worked does not exceed 5 to 10 pounds per square inch. The ratio of cylinder areas is about 9 to 1, the steam cylinder being the larger. Otherwise they are fitted up similar to pumps for general service. In some cases a hand power attachment is provided so that the pump can be worked by hand when the steam pressure is down.

PRESSURE PUMPS.

70. Pressure pumps are designed especially for use in connection with hydraulic lifts, cranes, cotton presses, testing machines, hydraulic machine tools of all kinds, and hydraulic presses, also for oil pipe lines, mining purposes, and such services as require the delivery of liquids under very heavy pressure. These pumps are invariably of the outside-packed plunger type and generally have four single-acting plungers working in the ends of the water cylinders, the latter having a central partition. The water valves are contained in small chambers capable of resisting very heavy pressures and ingeniously arranged for ready access. All materials used in the construction of the water end must be first class and suitable to the pressure used, which ranges from 750 pounds per square inch to 1,500 pounds per square inch. The water ends of these pumps are frequently made of hard, close-grained composition for medium pressures, and of steel castings for the heavier pressures.

MINE PUMPS.

71. Perhaps no other class of pump requires as much experience and skill to select as the mine pump. The reason for this is the wide variations in service, conditions of operation, head or pressure to be worked against, and the destructive nature of the water to be pumped. Nearly all the pumps at present installed are placed entirely below the surface. In former times the Cornish, or bull, pump was the favorite, but it is today abandoned for the more compact

and less expensive modern mine pump. The water end of the modern high-pressure mine pump may be described as having outside-packed plungers; strong circular valve pots independent of one another, but bolted to the working chamber, to the suction and delivery pipes, and to one another. Frequently the whole inside of the water end of the pump, from the suction nozzle to the discharge flange, is lined with wood, lead, or some other acid-resisting substance. Sometimes the entire water end is made of an acid-resisting bronze. Unless the service is light the outside-packed plunger pump is recommended for mine work; the valves should be preferably metallic valves in separate pots or chambers. Whether the pump shall be simple, compound, or triple expansion depends much on the price of fuel. In the anthracite coal regions the compound mine pump is now very common for sizes as small as 1,000,000 gallons in 24 hours, and they are invariably compounded for larger sizes, while the triple-expansion direct-acting pump is found in several of the mines.

72. Compound crank-and-flywheel high-duty pumps using the steam expansively have but recently been installed in the coal mines; in the iron and copper mines, where the cost of fuel is very high, the highest types of pumping engines have long been used.

73. When the larger types of high-duty pumps are used, the mine workings are generally so arranged that all the water runs to one large basin or sump near which a chamber of sufficient size is cut to contain the pump, which is surrounded and protected by suitable devices to maintain it in a high state of efficiency.

74. In many mines, strength and simplicity are the controlling elements in selection, for the reasons that many mines are compelled to use a large number of medium sized pumps and, for commercial reasons, use only one man whose business it is to make the rounds of the various pumps, giving each one but a few minutes' attention in a day. They generally have to stand rough usage, and the water pumped

is of such a corrosive quality that repeated renewals of parts of the water end are absolutely necessary. After heavy rains or other causes of flooding, the pumps are often required to run for days completely submerged and must pump both themselves and the mine dry. It can be readily seen that a pump for such service must be strong, simple, ready of access, and all of its parts of such construction that they can be readily taken apart or renewed.

SINKING PUMPS.

75. Sinking pumps are used in sinking or deepening mine shafts. There is little choice in their selection; generally speaking, they should be simple, strong, and capable of working in any position. The valves should be of the simplest possible construction and accessible for renewal with a minimum of labor and time. The valve motion should be simple and protected from dirt and drippings. They are regularly built single cylinder and duplex and are steam or electrically driven. With electric sinking pumps the protection of the electrical parts must be very complete.

BALLAST AND WRECKING PUMPS.

76. Ballast and **wrecking pumps** are principally confined to the marine service. The ballast pump is used on steamers having an extensive system of water ballast; also, for handling petroleum in bulk on oil-tank steamers. It is distinctively a special pump. The wrecking pump has a somewhat wider sphere. As its name implies, it is used principally by wrecking companies on the Atlantic and Pacific coasts and along the Lakes and is constructed with particular reference to reliability, portability, and general efficiency. It is well adapted to other services requiring the delivery of large volumes of water within the range of lift by suction. It has no forcing power, the water being merely delivered over the top of the pump, and it is single-acting, the water piston being fitted with valves. It is a

very light form of pump in proportion to the work it will do, is simple, durable, and not liable to derangement or breakage. It is also well adapted to drainage and irrigating purposes.

DEEP-WELL PUMPS.

77. Deep-well pumps, like sinking pumps, give little field for choice except in the pump-driving mechanism, which is as varied as the agent available to operate them, the principal agents being steam, electricity, gas, and windmills. The pump is usually a lifting pump having a bucket packed with numerous hydraulic leathers and working within the casing; it is usually given a very long stroke. These pumps do not handle gritty water successfully. Probably the best practical solution of the deep-well pump problem will be found in the air lift, which in principle and operation is quite simple.

SEWAGE PUMPS.

78. Sewage pumps are built in various types. When the lift is low, which condition is most common in sewage disposal, the centrifugal pump is the cheapest to install, but when economy and efficiency are important factors, the centrifugal pump must give place to the more expensive but more efficient reciprocating pump. Probably the largest single pumping engine ever constructed is the sewage pump for the city of Boston, which has a capacity of 70,000,000 gallons in 24 hours.

79. It will readily be seen that the selection of a type of sewage engine will depend much on the capacity of the installation and the price of fuel delivered at the station. The principal characteristics of the sewage engine are in the valves, which must be provided with very large ports to allow fairly large objects to pass through the pump without obstructing its valves. The valves are frequently made in the form of large leather-faced doors or flap valves, giving

nearly the full area of the pipe. The sewage pump does not differ in other respects from pumps for general service.

POWER PUMPS.

80. Power pumps are among the oldest styles of pumps, and may be developed by driving any type of reciprocating pump by other means than the use of a directly attached steam, gas, or air cylinder. Power pumps are very often geared or belted and with the increasing application of electricity the electric power pump is coming into more extensive use.

81. Power pumps may be used for any service and are frequently found in municipal water works, being often driven by a turbine or a Pelton waterwheel. In large electric-lighting, heat, and power plants, the power pump is much used for boiler feeding; in this case the pumps are usually triplex, giving a steady flow of water, and are driven by electric motors, the current being furnished by the main generators. This is probably the most efficient and economical boiler feeder that has been developed.

82. The power pump is used quite extensively in the mines. An electric motor being the driver, the system admits of many various sized pumps being placed at the different sumps throughout the mines and driven by one large and economical generating unit at the surface.

83. The selection of a power pump in preference to other types depends on conditions that, to some extent, may be gathered from the above applications; the choice, however, depends much on the kind of power available to run the machine. Where water-power is available, either for gearing directly to the pump or for generating electric current to drive the pump at a remote distance, the power pump may advantageously be chosen. It should be remembered in this connection that a steam pump should be installed to take

care of the feedwater when the main engines are stopped and no current is available for driving the power pump.

84. In private houses, hotels, office and public buildings the electric-power pump is a favorite, and to avoid the noise of gears the reduction in speed is made by friction drives of various types; rawhide gearing is also used to some extent. The construction of the water end of power pumps does not differ from other pump constructions for the same service.

MUNICIPAL PUMPING ENGINES.

85. While the **municipal pumping engine** may be of any size and capacity, and while some of the pumps already discussed, as the general-service and power pump, may be, and are, frequently used, the term usually implies the highest type of pumping engine that can be constructed as regards economy and efficiency. The refinement is more exacting as the capacity of the pump increases. For small municipal pumping engines, say of 2,000,000 to 5,000,000 gallons capacity in 24 hours, the compound and triple-expansion direct-acting engine is used, the degree of expansion depending on the price of fuel and the capital available for the investment. For installations of from 5,000,000 to 20,000,000 gallons capacity, the high-duty direct-acting engine, that is, the direct-acting engine with high-duty attachment, and the crank-and-flywheel engine are rivals for the installation; while for large municipal pumping engines above 20,000,000 gallons capacity in 24 hours, the vertical triple-expansion condensing three-crank single-acting, or differential, plunger beam type may be said to have no equal. With the latter type of engine a duty of 160,000,000 foot-pounds of work per 1,000 pounds of steam used by the engine is now common. Steam pressures of 175 pounds are common, while the number of expansions are as high as 22 to 26, and every reasonable device known in the art of steam engineering is used to the end of breaking records in securing a high duty.

VACUUM PUMPS.

86. Vacuum pumps are chiefly used in connection with jet condensers and siphon condensers. A vacuum pump is in reality an air pump, it being used for pumping air out of closed vessels. There are two general types of vacuum pumps, which are **dry vacuum pumps,** or pumps that handle air only, and **wet vacuum pumps,** or pumps that handle both air and water. Vacuum pumps are also used in some manufacturing operations where a high degree of vacuum is required, being used in connection with the vacuum pans found in sugar houses, with glycerin pans, etc.

RELATIVE MERITS OF DIRECT-ACTING AND CRANK-AND-FLYWHEEL PUMPS.

87. The relative merits of the two types of machine for a particular size, other conditions being equal, are such that it is a very difficult matter to decide which type is superior. For pumping small quantities of water, say up to 700 gallons per minute, and in localities where coal is not expensive, the direct-acting pump, either simple or compound, should prove a good investment. The objection to the direct-acting pump for large sizes is its waste of steam as compared with the crank-and-flywheel pump; it has an additional objection that is sometimes argued against it, which is *short-stroking*. This defect reduces its economical performance in that it requires some steam to fill up the space due to the incomplete stroke, but since the incomplete stroke is due to too high a compression, the compressed steam must have nearly filled the space before fresh steam was admitted, so that the loss is not so very great after all. Short-stroking reduces the capacity of the machine somewhat. In the common types of direct-acting pumps, the steam is not worked expansively; in compound and triple-expansion pumps, some degree of expansion is obtained, usually a little more than the ratio of low-pressure cylinder to high-pressure cylinder. By making the reciprocating parts

heavy and running the pump at some fixed minimum speed, an early cut-off can be effected in the high-pressure cylinder, the balance of the stroke being completed by the inertia of the reciprocating parts; in this way an increased degree of expansion is possible.

Another method of securing a considerable degree of expansion in the direct-acting pump is by means of the high-duty attachment. With the same degree of safety the speed of the direct-acting pump is very much less than is possible with the crank-and-flywheel pump. The direct-acting pump in which any attempt is made at economy will occupy quite as much space as the crank-and-flywheel pump of the same capacity, but the direct-acting pump is lower in first cost.

88. Probably the most objectionable feature of the crank-and-flywheel pump, which is an inherent one, is that the velocity of discharge varies throughout the stroke. This is due to the fact that while the flywheel rotates at a uniform speed, the pistons and plungers move with a variable speed, varying from zero at the beginning of the stroke to the maximum speed near mid-stroke and then decreasing to zero at the end of the stroke. This variation in velocity produces shocks, and hence requires the water end of a flywheel pump to be of heavier construction than a similar end for a direct-acting pump. The valve area of a flywheel pump requires to be considerably larger than for a direct-acting pump, not only because of its capacity for higher speeds, but also because the velocity of the plunger, when the connecting-rod is at right angles to the crank arm, is somewhat in excess of 1.57 times the mean velocity of the plunger. In addition to the greater valve area and strength required in flywheel pumps, it is necessary to use some means to reduce the shocks to the mechanism and parts of the pump. This is accomplished by providing large air chambers, preferably one over each deck for high pressures; for very high pressures and long columns of water, an alleviator is necessary.

89. The main advantage of the crank-and-flywheel pump is its economy, which, in turn, is due to the fact that the

steam may be expanded to any permissible degree; it also readily admits of all the refinements known of securing high-duty performance, and with a proper arrangement of details, it can be made quite as safe as ordinary machines. For extreme high duties the crank-and-flywheel pump is always chosen, and to reduce the shocks due to a variable discharge a favorite type is the three-crank machine. The combined delivery from three plungers is tolerably uniform and the arrangement readily lends itself to the extremely economical triple-expansion condensing engine.

90. The crank-and-flywheel engine is more expensive than the direct-acting machine, and when high degrees of expansion are used occupies considerably more room. It is generally more complicated, but is more accessible, except in such cases as where an effort is made to minimize space, when by making the engine back-acting it is liable to become quite inaccessible.

91. The piston speed of direct-acting pumps rarely exceeds 100 feet per minute, while the piston speed of crank-and-flywheel pumps is commonly 300 feet and sometimes 400 feet. With pumps of the controlled-valve type, piston speeds of 560 feet are reached. This difference in the piston speed of the direct-acting and crank-and-flywheel pumps shows that they must be compared on the basis of water delivered rather than on the relative size of similar parts.

92. Even for very small sizes, the crank pump is sure in its action and is not liable to get out of order; this cannot be claimed for some of the single-cylinder direct-acting pumps having steam-thrown valves. The crank pump is limited as to its slowest speed, however, since the speed must be sufficient to store energy enough in the flywheel to carry the crank over the dead centers. This objection can be overcome to a great extent by using the by-pass, which allows part of the water to be returned to the suction, thus decreasing the work on the pump.

INDEX

NOTE.—All items in this index refer first to the section (see the Preface) and then to the page of the section. Thus, "Accumulator 34 26" means that accumulator will be found on page 26 of section 34.

C

D

G

N

O

P

S

T